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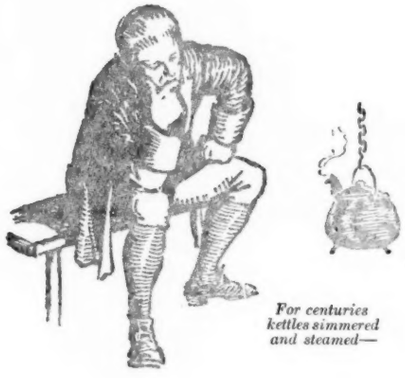
The Thinker

By R. T. Strohm

THIS land is fast filling with chaps who are willing
To barter their bodies where labor is sold;
The beefy and brawny, the stunted and scrawny,
Work blindly at doing whatever they're told.
The roughneck, rascallion, Hun, Pole and Italian
Dig shoulder to shoulder with negro and Chink.
Ill-paid for the hustle and straining of muscle,
And never expected to reason or think.

The beefy and brawny work blindly—

FOR centuries, kettles of stoneware and metals
On hearthstone and trivet had simmered and steamed,
While savory vapors in sinuous capers
Had wantonly wasted a power undreamed;
Till Watt chanced to wonder what forces were under
The covers to cause them to clatter and clink,
And history's pages through all coming ages
Will give him due credit for pausing to think.



For centuries kettles simmered and steamed—

CRAB, pippin and sweeting, too ripe for good eating,
Had fallen each autumn since Cain was a kid,
But mankind was lazy or too lackadaisy
To try to find out why they dropped as they did.
So, Old Gravitation had no explanation
Till Newton discovered the long-missing links,
Thus clinching securely the honors that surely
Become the reward of the fellow who thinks.

Crab, pippin and sweeting had fallen since Cain was a kid—

THERE'S plenty of room in the head of each human
To suit a good, workable outfit of brains,
But yet the world suffers from thousands of duffers
Who don't know enough to come in when it rains.
So, when they talk plainly or grumble profanely
And say that Success is a treacherous minx,
Just step up behind them and gently remind them
There's room at the top for the fellow who thinks.

New Brunswick Boiler Explosion

SYNOPSIS—Additional data relating to the cause of the boiler explosion. The rear sheet was overheated, but no other portion of the boiler was burned. The direct cause of the overheated plate that tore apart has not been determined.

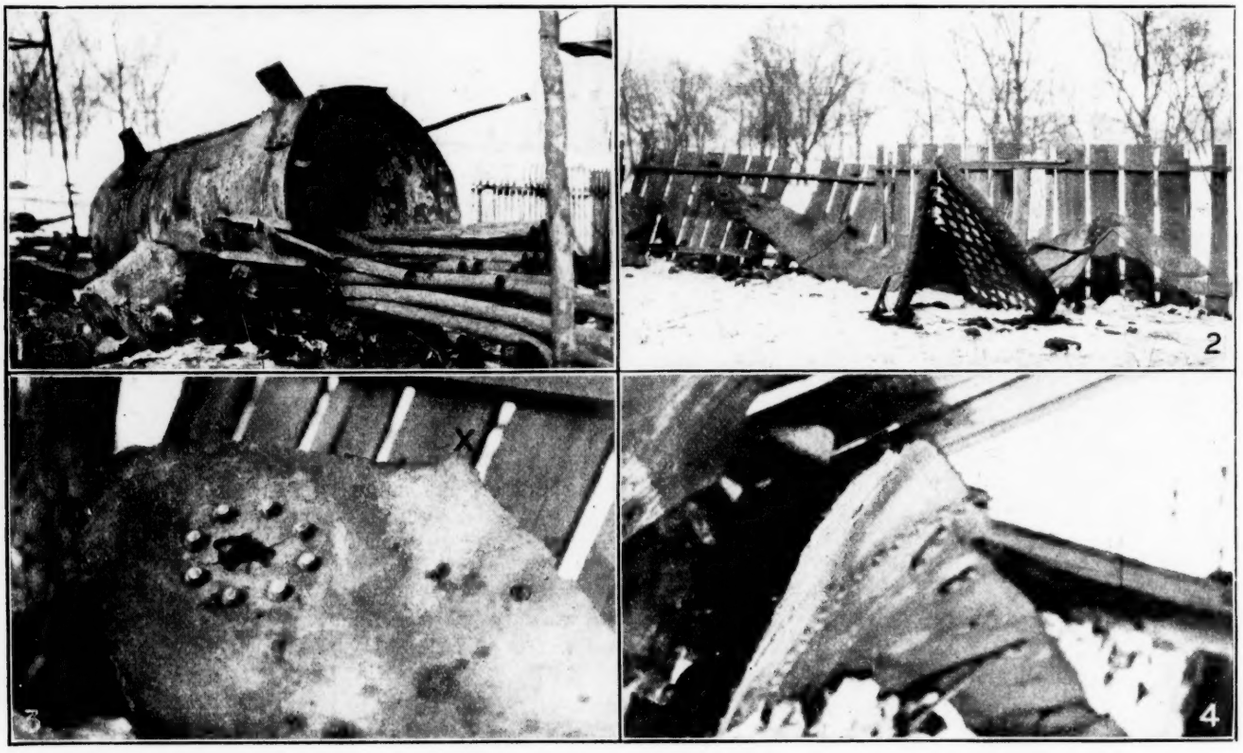
The disastrous result of the boiler explosion at the plant of the Howe Rubber Co., New Brunswick, N. J., a report of which was published on page 237 of the Feb. 15 issue of *Power,* is shown in the accompanying illustrations, which were received too late to use in that issue.

As stated, the return-tubular boiler was 60 in. diameter and 14 ft. long and operated with 70 lb. pressure. It exploded Wednesday morning, Feb. 2, killing two men and causing a loss of $50,000.

Investigation has brought out the fact that the boiler was repaired on Jan. 30, the Sunday before the explosion. The repair consisted of putting on a pad, or flange, for a blowoff connection in the bottom of the rear sheet. The old connection was in the back head, close to the bottom. The change was made to take care of a radial crack, shown in Fig. 3, and as an alternative to putting on a good-sized patch.

It was stated in the previous article that the immediate cause of the explosion was overheating around the blowoff outlet. The tubes showed some old lime scale, but the water now used contains no scale-forming matter. No other portion of the boiler showed overheating. It is a mystery, therefore, as to where the material came from that caused the plate to overheat. It is evident that something settled on the plate around the blowoff, perhaps accumulating and getting baked together in the two days sufficiently to keep the water away from the plate. What this material was will probably never be known.

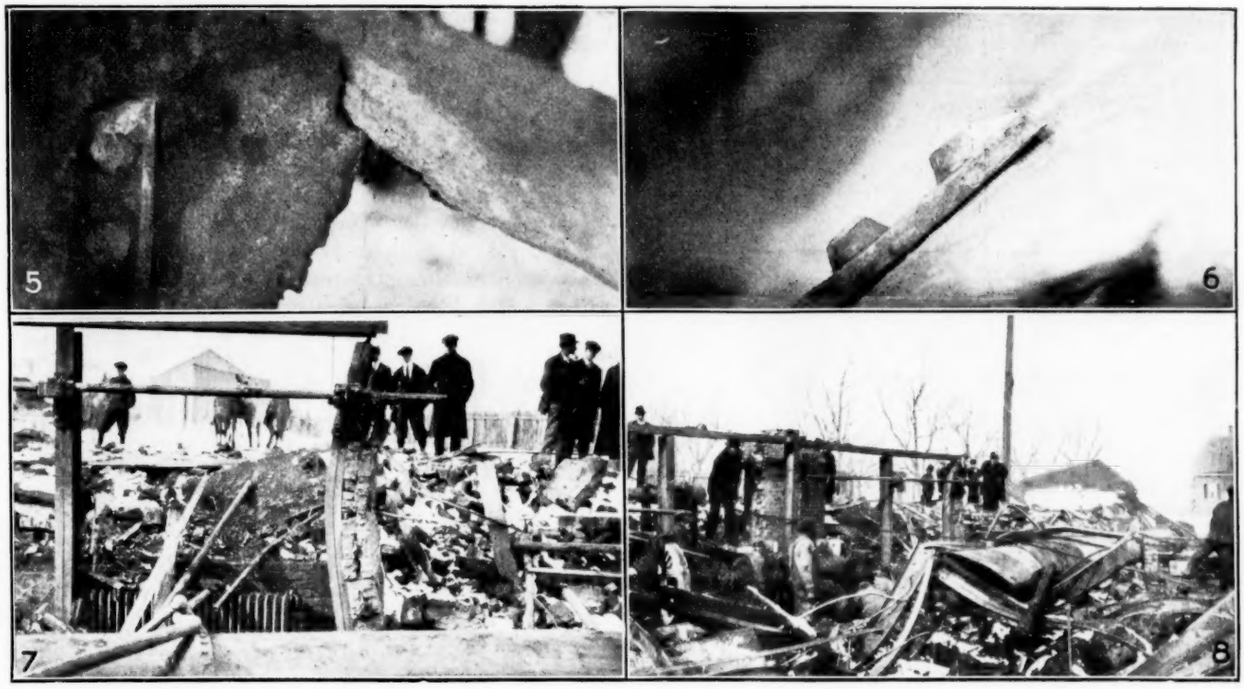
A study of the accompanying illustrations is of interest, as they show that the plate, instead of failing at the longitudinal seam, fractured at a point about midway of the sheet, between the blowoff flange and the rear girth seam, as shown at *A*, Fig. 9. A plan of the boiler room, Fig. 10, shows the storeroom at the left and the factory at the back end of the boiler. The crack *AB* indicates where the rear plate tore, and the figures indicated at *D* and *C* show the direction and distance the boiler and rear header and a portion of the rear sheet were blown.



FIGS. 1 TO 4. DETAILS OF THE EXPLODED BOILER AND TORN REAR SHEET

Fig. 1—Shell of the boiler where it was blown. Fig. 2—Rear head, with portion of attached sheet. Fig. 3—Details of new blowoff outlet. Tear started at the point X. Fig. 4—Corner X of Fig. 3, showing burnt portion white

The body of the boiler, Fig. 1, is shown where it fell, 10 ft. from its original site. The shell was slightly dented. The rear head, with the attached piece of the rear sheet, is shown in Fig. 2, having been blown a distance of 60 ft. The left end is shown enlarged in Fig. 3, where the details of the blowoff outlet disclose the old crack. The burned surface shows white in the illustration, but the original plate showed blue, indicating beyond a doubt that it was overheated. The plate started to tear at the point *X*. Another view of the burned plate is shown in Fig. 4.



FIGS. 5 TO 8. MORE DETAILS OF THE EXPLODED BOILER. ALSO SHOWING THE PLANT DEBRIS

Fig. 5—The point X indicates where the sheet started to tear and where the first leak started. Fig. 6—Brace with scale adhering even with the rivet head. Fig. 7—Wreckage, showing bent column at the rear of the boiler. Fig. 8—Wreckage from far end of the factory; stack in the center

The right end of the plate, Fig. 2, is shown enlarged in Fig. 5. The crack at X may have been the initial leak heard before the plate failed altogether. The worst-burned portion is shown at the right of the crack. One

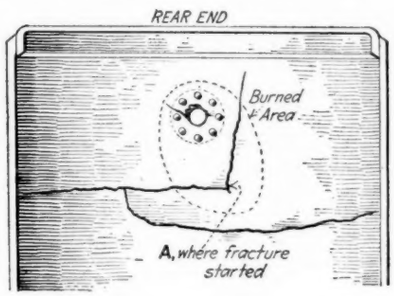


FIG. 9. DETAIL OF BURNED AREA AND CRACKS OF THE REAR SHEET

of the braces, with adhering scale of a thickness equal to the rivet head, is shown in Fig. 6.

Two general views of the wrecked plant are shown in Figs. 7 and 8. In the former the bent column may be seen at the rear of the boiler setting, the factory room being at the left. The steel stack in the foreground fell on the roof of the factory. Wreckage from the far end of the factory, with the top of the stack in the foreground, is shown in Fig. 8. The factory hand killed and the one most seriously injured were in the rubber mill by the machine shown.

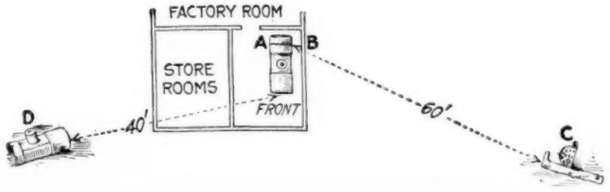


FIG. 10. SHOWING PLAN OF BOILER ROOM AND THE DIRECTION IN WHICH THE BOILER PARTS WERE BLOWN

The total destruction of the factory was mainly caused by fire which broke out immediately following the explosion and was probably started by the hot coals from the furnace being thrown to several parts of the factory.

As is usual in boiler explosions, various causes were given according to newspaper accounts. One was that the water was low, another that the boiler was defective. Examination of the boiler plates gives evidence that neither of these was the cause of the disaster.

As the water recently used in the boiler did not contain scale-forming material, it is not probable that newly formed scale was responsible. There is, however, the possibility that old scale had become loosened from the tubes and shell and collected at the burned area around the newly made blowoff outlet. It is also possible that a workman carelessly left a jumper in the boiler or a bundle of rags used to kneel on while riveting the blowoff flange in place. If such was the case and loose scale accumulated over them, the overheating of that portion of the boiler in the vicinity of the blowoff would be accounted for.

The sum and substance of the matter is that a collection of some foreign matter collected during the two days the boiler was operated after the repairs were made—just what it was is a mystery.

The lesson to be learned is that every boiler, after workmen have been engaged on the interior, should be thoroughly inspected by a competent, careful man to see that every tool, piece of waste, loose scale, etc., have been removed before the boiler is filled and put under steam pressure. Trusting to the vigilance of the other fellow is not good engineering unless he is a man of responsibility.

Bearing-Design Constants--II

SYNOPSIS—Independent sets of design constants that have been developed in the various branches of machine design are coördinated by finding the common law of friction underlying them.

Having fixed upon the probable value of the friction coefficient, it is now possible to derive the bearing design constants on the basis of equation (8). The heating effect that may be expected in a bearing operating at a pressure less than the critical may be found by substituting in equation (1) the value of the coefficient μ, thus,

$$V_{lim}P = \frac{H_o t_h}{\mu} = 2.5 H_o t_h \sqrt{P T_o} \qquad (11)$$

where

V_{lim} = Maximum velocity allowable for given pressure and temperature conditions.

This equation is dependent upon an ample supply of suitable lubricant, conforming to the specified rate of feed of about 350,000 sq.ft. of surface per gallon of oil.

Transposing the equation, the limiting velocity permissible under assumed conditions of radiation becomes equal to

$$V_{lim} = 2.5 H_o t_h \sqrt{\frac{T_o}{P}} \qquad (12)$$

Should the actual velocity V lie below the permissible value V_{lim}, it may be considered that the bearing is reasonably safe and that this provides a sufficiently liberal margin to allow for partial scoring, lack of fit and other ordinary contingencies such as may be expected to arise under competent supervision.

Heat Absorption and Offtake

Analyzing equation (12), it will be seen that the allowable velocity increases with the specified radiating capacity H_o, showing that the bearings for high-speed work, if not water-cooled, should be of massive construction to provide liberally for the heat offtake. The velocity may also be extended in proportion to the rise in temperature head, hence it is usually found expedient to maintain a relatively high temperature in high-speed bearings. This same factor is again embodied in T_o, since the friction coefficient is reduced at high temperature. Accordingly, in bearings of this type the best results may be obtained by working at a relatively high temperature and low pressure. This combination is permissible because the pressure will in any event lie below the critical limit set by equation (9), and hence the oil may be allowed to become quite fluid without detrimental results.

A temperature rise of $t_h = 100$ deg. F., with respect to an atmospheric temperature of about $t_o = 100$ deg. F., represents the limit that should not be exceeded, while for design purposes a temperature rise of about 75 deg. F. at about 90 deg. F. atmospheric temperature is a good basis for naturally cooled high-speed bearings.

In case the radiation is increased by artificial cooling, the velocity may be proportionally extended. This also permits the use of heavier pressures in which case it is best to restrict the temperature rise to about 50 deg. F. and work with the more viscous oil in the manner of a main engine bearing.

In outboard bearings, which usually run at relatively slower speeds and heavier pressures than main bearings, it is advisable to reduce the temperature rise to, say, about 40 to 45 deg. F. Some designers still deem it expedient to stay under the temperature limits set, but while such practice does provide for a somewhat greater safety factor, it unnecessarily increases friction losses.

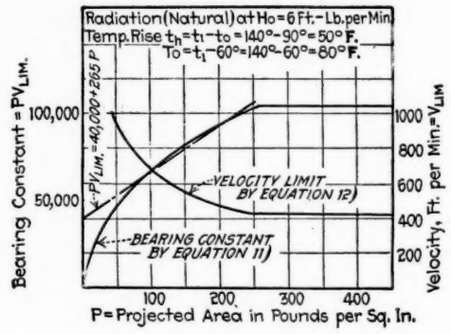

Radiation (Natural) at Ho = 6 Ft.-Lb. per Min.
Temp. Rise th = t1 - to = 140° - 90° = 50° F.
To = t1 - 60° = 140° - 60° = 80° F.
Bearing Constant = PVLIM.
100,000
50,000
PVLIM. = 40,000 + 265 P
VELOCITY LIMIT BY EQUATION 12)
BEARING CONSTANT BY EQUATION 11)
1000
800
600
400
200
Velocity, Ft. per Min. = VLIM
100
200
300
400
P = Projected Area in Pounds per Sq. In.

FIG. 7. LIMITS OF VELOCITY AND PRESSURE BY EQUATIONS 11 AND 12

As based upon the friction data discussed, the following equation may be taken as a more rational form for the bearing constant:

$$\text{(Design Constant)} \quad V_{lim}\sqrt{P} = 2.5 H_o t_h \sqrt{T_o} \qquad (13)$$

This formula applies primarily to a steel journal running in a rigid babbitt-lined bearing shell.

The results given by equation (11) have been plotted in Fig. 7 for limits of velocity and pressure corresponding to main-bearing practice, the natural radiation factor H_o being taken at 6 ft.-lb. per min. at a temperature rise of 50 deg. F., with respect to an atmospheric temperature of 90 deg. F. and the corresponding velocity limits as fixed by equation (12) are likewise plotted.

Fig. 8 shows similar relations as deduced for high-speed bearing practice, H_o being taken at 4 ft.-lb. and the temperature rise at t_h at 75 deg. F.

When applying any of the design constants as given herein, it is to be remembered that they represent the maximum allowable values when running at the limiting velocity V_{lim} as fixed by equation (12). When the actual velocity V lies below the maximum allowable velocity V_{lim}, as is generally the case, it becomes necessary to reduce the given constant in the ratio of $V \div V_{lim}$, and provided the critical pressure P_c is not exceeded, the temperature rise will drop in approximately the same proportion.

Modification in Design

Under certain design limitations, the pressure and velocity of the bearing may, however, be such that the desired temperature rise cannot be reached, in which case it becomes necessary to interpolate until the temperature rise and friction coefficient comply with the conditions set forth in equation (1).

Referring to Fig. 7, it will be seen that for usual bearing pressures of from 100 to 250 lb. per sq.in. of

projected area, the curve can be quite closely approximated by the following equation:

(Engine Bearings) $PV_{lim} = 40{,}000 + 265P$ (14)

This formula applies only when $H_o = 6$ ft.-lb. and when the temperature head $t_h = 50$ deg. F., while for different conditions it is best to use the fundamental equations (12) and (13).

The velocity limits shown in Fig. 7 refer to natural cooling, while for water-cooled bearings the velocity allowable would be raised in direct proportion to the cooling effect. It will be apparent that for usual speeds in engine bearings, lying between 200 and 500 ft. per min. artificial cooling is unnecessary, especially so when the construction is sufficiently massive to provide for a generous conduction factor H_o.

When applying these constants to nonadjustable bearings such as are commonly used for single-acting vertical gas or oil engines, the bearing pressure P should be taken as the dead-load pressure plus the average pressure occurring during any one cycle; that is, proportional to the mean between the average pressure during the expansion and compression strokes.

Comparison of Results

Equation (14) is in a form that allows the writer's deductions to be conveniently compared with the formulas generally used for design purposes.

Without going into details, it may be stated that the proposed equations have been checked and appear to be in satisfactory agreement with a number of design constants heretofore used. This fact therefore furnishes further proof that the coefficient of friction attained in commercial service may be expected to approach the values fixed by equation (8).

The writer has found equation (13) to check very well for a number of babbitt-lined outboard bearings. The results also apply to quarter-box main bearings for horizontal heavy-duty engines, when the projected area of the bottom block is assumed to carry all of the dead, or downward load, while that of the cheek-block is assumed to take the average cylinder load in the manner defined.

The P V constant for either block considered separately, should not exceed the limit set by equation (14). In addition, the temperature rise of the bearing, considered as a whole, should be checked for heating by taking the total pressure on each of the surfaces independently and multiplying by the respective friction coefficient; the gross lost work thus found should not exceed the radiation limit prescribed by the factor H_o, as measured on the basis of projected journal area. When checking the side cheek, the value of the coefficient μ as given by equation (8) should be increased to at least $\frac{3}{2}\mu$ in order to make the total temperature rise of the bearing conform to the results attained in practice. The larger part of this extra friction is no doubt to be ascribed to conditions similar to those existing in the cheek-block and the Thurston testing machine previously commented upon.

Allowable Pressure Limits

In applying the foregoing design constants to slow-speed bearings, the value given as the allowable pressure may readily become unduly large, while experience shows that certain well-defined limits must not in any case be exceeded.

For ordinary or soft babbitt-lined bearings, it is found unsafe to carry a dead-load pressure much beyond 300 lb. per sq.in. of projected area, since otherwise the babbitt is likely to be squeezed out of the shell. Genuine babbitt metal, because of its greater elastic strength, can carry heavier pressures, but a point is soon reached where such considerations as excessive wear and the like put a limit upon further pressure increase. Furthermore, the uneven distribution of heavy pressures and the tendency to score the wearing surface are additional factors that restrict bearing pressures.

Experiments made by H. B. McDermid[11] on pressure limits for babbitt metal of varying composition show that soft, or lead, babbitt may be expected to fail at from 300 to 500 lb. per sq.in.

Influence of Metal and Service

The fact that these higher pressures exceed the critical pressure P_c may also be a contributing factor in setting a limit upon the allowable bearing pressure. Referring to Fig. 7, it will be seen that the P V constant for engine bearings reaches a fixed value beyond a pressure of about 250 lb. per sq.in. and a velocity of 420 ft. per min. The critical pressure will therefore be exceeded whenever the velocity is less than 420 ft. for a pressure of 250 lb. or more.

For railway brasses running on steel journals, the dead-load pressure is usually restricted to about 400 lb. per sq.in. of projected area. For intermittent pressures,

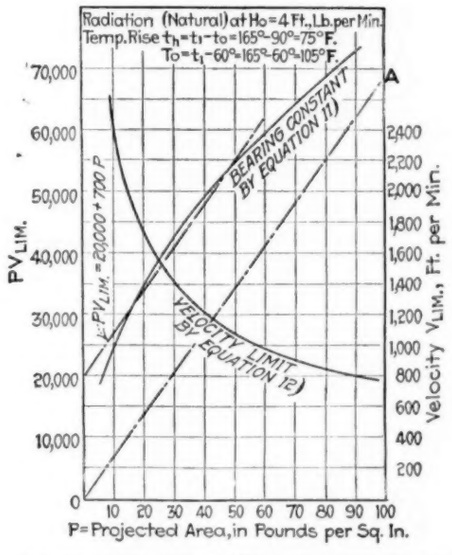


FIG. 8. ALLOWABLE VELOCITY AT LOWER PRESSURE AND GREATER TEMPERATURE RISE THAN IN FIG. 7

such as occur in crankpins, the limit of pressure is just about four times that allowable for dead loads.

Referring to Fig. 8, showing the velocity and pressure limits for high-speed bearings, the curves clearly indicate the necessity of decreasing the pressure as the speed increases. It will also be seen that even with bearings of heavy construction the velocity is restricted to about 2,500 ft. per min. when radiating into the atmosphere. To be able to use higher velocities or greater pressure, the bearing must be artificially cooled to avoid excessive heating.

[11] "Investigation of Bearing Metals"; "Power," Jan. 10, 1911.

The results that would follow had the high-speed bearing constants been based upon Lasche's values for the coefficient μ as given by equation (6) are shown by the dotted straight line $O\ A$, Fig. 8. The corresponding velocity limit would be a constant for varying pressures, and its value lies far below that customarily used in high-speed bearing practice.

As is further shown by Fig. 8, the $P\ V$ constant as determined for heavy high-speed ring-oiler bearings, having an assumed radiating capacity of $H_o = 4$ ft.-lb. and a temperature rise of $t_h = 75$ deg. F., may be approximately expressed by the following relation:

(High-speed Bearings) $$P V_{lim} = 20{,}000 + 700P \quad (15)$$

This formula applies to $P = 10$ to 60 lb. per sq.in. and $V_{lim} = 1{,}000$ to 2,500 ft. per min.

In case of lighter bearings having less radiating power, the foregoing constants and velocity limits must be proportionally reduced. Similarly, a restricted temperature rise or lack of rigidity and like considerations may dictate the use of lower bearing constants.

The length of journals for high-speed bearings, such as are covered by equation (15), are commonly made equal to about three times the journal diameter. In high-pressure bearings as covered by equation (14) the load is less likely to be uniformly distributed and the length is generally made equal to about 1½ to 2 times the journal diameter, 1¾ diameters being a good average proportion. Equation (15) is found to check for the ring-oiler bearings such as are generally used for fans, generators, turbines and like machinery.

Equations Checked by Tests

A good check is afforded by some motor-bearing tests made by A. M. Bennett[12] on a 1¾x5¼ in. journal running at 4,000 r.p.m., or 1,830 ft. per min. Various types of bearings were experimented with, but only the standard ball-seated babbitt-lined ring-oiler bearing. The average temperature rise was found to be 71 deg. F. at an average pressure of about 25 lb. per sq.in. of projected area, the corresponding $P\ V$ constant being 46,000. A check for the radiation constant shows this to be approximately $H_o = 4$ ft.-lb. and hence according to Fig. 8, the $P\ V$ value becomes equal to about 38,000 at $V_{lim} = 1{,}780$ ft. per min. This comparison shows that equation (15) restricts the bearing constant to values below those found permissible in practice and furnishes further proof that the coefficient μ as given by the Lasche equation (5), is undoubtedly higher than that usually met with in the practical operation of low-pressure bearings.

Further evidence could readily be brought to bear upon this point, but the foregoing will no doubt be sufficient to substantiate the correctness of the general conclusions embodied in the design constants presented.

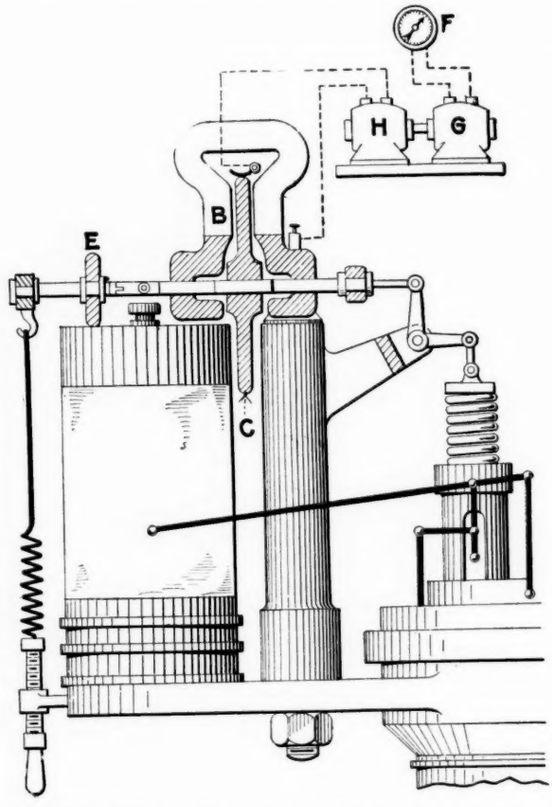
It is believed that a decided impetus would be given to scientific research if, in determining the coefficient of friction, all of the aspects outlined were more fully taken into account by investigators. Test results more nearly in keeping with practical experience would not only contribute materially to a more rational basis for bearing-design constants, but would also render valuable aid to machine designers in further reducing friction power waste.

[12] "Temperature Tests on Bearings"; "Power," June 17, 1913.

Continuous Engine Indicator

The device shown in the illustration, for continuously indicating the horsepower developed by a steam engine, has been patented by a Hungarian engineer, M. Korosy, according to the *Practical Engineer* (England).

The apparatus consists of a magneto the armature C of which is rotated by the friction wheel E, the distance of which from the axis of the drum is determined in an obvious manner by the pressure upon the piston of the indicator. The current generated by the armature in



INDICATOR AND ARRANGEMENT OF ELECTRICAL APPARATUS

passing between the pole pieces of the magnet B is transmitted to a motor H, and the voltage generated by the direct-connected generator G, indicated upon the voltmeter F, is translated into terms of horsepower. The greater the pressure exerted on the indicator piston the greater the radius of the circle in which the friction wheel E travels, hence the greater the speed of the armature and the greater the voltage or horsepower. The idea appears to be hardly practicable, and one wonders why the current generated by the armature cannot be measured directly, instead of being used to run a motor-generator to make current to be measured.

Viscosity Tests, so called, do not necessarily prove anything of the lubricating quality of an oil, since rosin or other viscous oils may constitute a large proportion of the whole and yet have no lubricating value.

Special Mixtures of Oil are necessary in some cases, but for light machinery using only small quantities, a mixture of 80 per cent. light mineral and 20 per cent. sperm is good and should not cost more than 30c. per gallon. It will not, however, stand heavy bearing pressure or form a film at slow speed. Its film thickness on metal is about 0.0002 in.

Parallel Operation of Direct-Current Generators--I

By Alan M. Bennett

SYNOPSIS—Discussion of the characteristics or properties of series, shunt and compound-wound generators insofar as they affect the operation of machines in parallel. This installment takes up the requirements which must be fulfilled particularly in the field control, for successful parallel operation. A later article will take up three wire generators and balancers.

On constant-potential systems, generators are connected in parallel when the demand for current on the system becomes greater than can be furnished by a single machine. The current from each machine as it is placed in circuit is added to that of the machines already in operation, the voltage remaining constant irrespective of the number running. The requirements for successful operation when paralleling in this manner are (1) that the incoming generator be thrown in circuit with the least possible disturbance to the system voltage and (2) that the generators divide the load in proportion to their capacities.

Influence of Characteristics of Different Types of Generators

Series generators are not commonly used on constant-potential circuits, because their voltage varies over a wide range with changes of load. Their operation will be considered merely to show its similarity to that of compound generators when run in parallel. Shunt generators are frequently run in parallel, and while their voltage varies to some extent with change of load, it is not subject to wide fluctuations, as in the case of series machines. Compound generators, however, on account of their self-regulating feature of maintaining a constant voltage even under extreme conditions of load, are in almost universal use on power and lighting systems.

The connections of shunt-wound generators are simple, and there is no particular difficulty in getting them to share the load properly, if started with the same terminal voltage and their speeds do not vary appreciably. It must be understood, however, that no generator while at rest can be switched into parallel with others that are running, without danger of short-circuit. The armature of the incoming machine in this case would be thrown directly across the line, and the current that would flow through it would be limited only by the resistance of the armature. It is necessary, therefore, first to bring the generator up to the voltage already existing on the system, before throwing it into parallel. As the machine is brought up to speed with the field rheostat cut in, its voltage gradually rises till full speed is reached. The excitation is then adjusted by means of the field rheostat until the voltage is the same as, or slightly higher than, that of the busbars, when the generator may be thrown into parallel by closing the main switch and again adjusting the voltage until it carries its proper share of the load. Fig. 1 shows diagrammatically the connections for two shunt generators in parallel, with the field of each connected directly across its armature. With this connection the field is independent of the outside circuit, and with a given rheostat setting, builds up as the generator comes up to speed.

Shunt Fields Across Busbars

Sometimes, however, the field, instead of being connected as in Fig. 1, is connected across the main busbars, as in Fig. 2. In this case it is necessary to have a switch in the field circuit, and this must be closed when the second machine is to be brought into operation. The fields by this means are thrown directly on the full line potential, and when the generator is started, its voltage builds up rapidly. When the first machine is started, the main switch must be closed as well as the shunt-field switch, in order to complete the circuit through the fields.

In throwing a generator out of circuit, resistance is cut into its field by means of the rheostat, until the ammeter shows practically no current being delivered to the line. The main switch can then be opened without flashing, and in the arrangement shown in Fig. 1 the generator is completely disconnected from the line. In Fig. 2, however, it is necessary to disconnect the field also, and as this is partly excited, to open its switch would cause a dangerous flash due to the self-induction of the field coils. The field should therefore be shunted with a non-inductive resistance, when the switch can be opened without flashing, the self-induced current from the field passing around through this resistance without danger.

Regulating Resistance in Field Circuit

It will be found that comparatively little resistance will have to be cut in by the rheostat to reduce the output to zero. It is necessary only to bring the internal voltage of the generator down to the point where it is the same as the voltage of the line, when the generator will cease to furnish current to the external circuit. This can be seen when it is considered that the internal, or total, voltage of the generator is greater than that of the line by the amount of the voltage drop in the armature ($I^2 = R$). Consequently the nearer the internal voltage approaches that of the line the less current is delivered, and when these two voltages become equal no current is delivered to the line. If the voltage be still further reduced so that it becomes less than that of the line, the generator will begin to draw current from the line and run as a motor, in which case the generated voltage becomes a counter-electromotive-force, tending to limit the current taken by the machine. It is, therefore, not desirable to introduce more resistance into the field circuit than is necessary to remove the load from the generator.

The fact that the voltage of a shunt generator drops with increasing load tends automatically to keep the division of load stable when running two of them in parallel. If one machine should suffer a voltage drop due to a drop in engine speed, it would be instantly relieved of part of its load, which would be taken by the second machine. The result would be an increase in voltage on the first machine due to the decrease in its load and a

decrease in voltage on the second machine due to its increased load, this condition tending automatically to pull the generators into proper working and adjust the load between them. The case in which one machine is over-powered by another is therefore uncommon and can only take place when its internal voltage falls below that of the line, which would call for considerable drop in

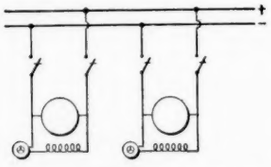


FIG. 1. SHUNT GENERATORS, FIELDS ACROSS ARMATURES

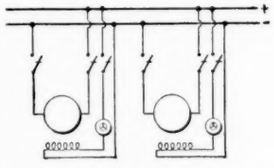


FIG. 2. SHUNT GENERATORS, FIELDS ACROSS BUSBARS

speed. Even should this occur, the direction of rotation would still remain the same; there would be danger, however, of an abnormal demand for current from the generator, with a consequent blowing of fuses.

Two compound generators possessing like characteristics and driven at the same speed will generate equal voltages when the excitation, or magnetizing effect produced by the series field, is the same in each, or when the current through these fields is the same. Under this condition the voltage drop in the two fields is alike. If two such generators were connected to the line in the manner shown for shunt machines and started with the same initial voltage and if, due to variation in speed, the voltage of one should drop slightly, its load would immediately fall off and be taken by the second machine. The decrease in current delivered by the first machine would cause a further decrease in its voltage, owing to the lessened field excitation, while the voltage of the second machine would increase, owing to increased excitation caused by the additional current through the series field. By this unbalancing of the voltages, the first machine would deliver still less current, and the second still more and this action would continue until the second machine was carrying the total load, the first being entirely robbed of its share.

Correcting Unbalancing Tendency

In order to correct this tendency to unbalance, it is necessary to raise the voltage of the machine that suffers a decrease in load, and this is done automatically by employing an equalizer connected to the series field of each machine at the end nearest the armature, thus placing the series fields in parallel, as shown in Fig. 3. The function of the equalizer is to keep the magnetizing effect of the series field the same in the two machines, its action being as follows:

As long as the two generators maintain equal voltages, the points c and d will be at the same potential and no current will flow between them through the equalizer, the full output of each machine passing through its series field. Should the voltage of generator A rise slightly, its current output will increase and there will be a proportionately greater drop in its series field. The point c will now be at a higher potential than d, and current from A will flow from c to d through the equalizer and thence to the positive busbar through the series field of generator B, increasing the excitation of this field and correspondingly raising the voltage of B. This action will continue until the points c and d have been brought to the same potential and the load is again properly divided. If, instead of increasing, the voltage of generator A had fallen, the reverse action would have taken place; c would be at a lower potential than d, and current would flow from d to c. Thus it is seen that current may be flowing through the equalizer in either direction or it may be carrying no current at all, depending on the momentary value of the voltages of the two machines, and that the current in the series field may be more or less than the armature current by the amount passing through the equalizer.

It is necessary that the cross-section of the equalizing connection be sufficiently large to prevent an appreciable drop in voltage along its length, otherwise its function will be interfered with and the weaker machine is likely to be overpowered and converted into a motor. Should reversal of polarity take place through the action of reversed current in the series field, rotation as a motor would be opposite to that as a generator and serious damage would result. In order, therefore, to insure proper action, the resistance of the equalizer must be low compared to that of the series fields, and this can usually be effected by making its cross-section the same as that of the main leads to the busbars.

It is usual with compound generators to connect the series field on the positive side of the generator, but the equalizer works as well when the series fields are on the negative side. It is necessary, however, that all machines operating in parallel have their fields similarly connected, all on the positive or all on the negative side.

Paralleling Different Size Generators

Generators of different capacity and make can be paralleled satisfactorily if the resistance of the series fields of the two machines be proportioned to the current designed to be carried by them.

The usual arrangement for connecting compound generators to the line is to run the equalizer and the lead from the series field to a double-pole switch and the lead from the opposite side to a single-pole switch. In starting up, all switches should be open. The generator is

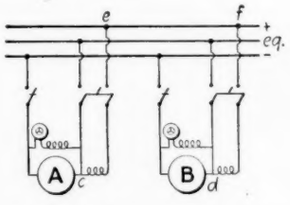


FIG. 3. COMPOUND GENERATORS, USUAL FIELD CONNECTIONS

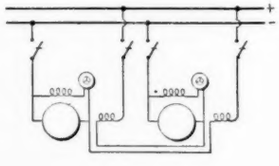


FIG. 4. COMPOUND GENERATORS, MUTUAL EXCITATION

brought up to speed and the voltage adjusted by the rheostat until it is approximately the same as that of the line. The double-pole switch is then closed and the voltage again adjusted until it is the same as, or slightly higher than, that of the line, when the switch in the other main can be closed, putting the machine in circuit. By closing the double-pole switch first, the series fields of the second generator receive current from the line before the generator is thrown in circuit, and the voltage builds up more rapidly. There is also less likelihood of voltage disturbance on the line if the generator is equalized before

being paralleled, and a method sometimes used to insure this end is to connect the voltmeter by which the generator is brought up to voltage so that it is not in circuit until the equalizer switch is closed.

An alternative arrangement of switches much used is to connect all three leads from the generator to a three-pole switch, which is closed after the generator has been brought up to voltage. The disadvantage of this arrangement is that the generator is not equalized before it is connected to the circuit.

To throw a generator out of circuit, its load is reduced by cutting in resistance in the same manner as described for shunt generators, and the line switches opened, in the case of the single- and double-pole switches, the latter being opened last.

When generators are first operated in parallel after installation, trouble is often experienced in getting a proportionate division of load between them. In such cases it will usually be found that the resistance of the series fields and their connections to the busbars are not correctly proportioned to the current they are designed to carry. Referring to Fig. 3, it is necessary that the voltage drop from *c* to *e* be the same as from *d* to *f*, when the machines are carrying their proportionate share of the load; and if this is not the case resistance must be added to one or the other of the two circuits until they have approximately the same drop. Usually, an extra length of cable in the circuit that is deficient in resistance will produce the desired results. It is sometimes attempted to correct such trouble by an adjustment of the series shunts on the generators, but beyond affecting the compounding of the machines nothing can be gained by this method, for the series shunts themselves are in parallel, and a change in one affects both generators.

Ammeters and circuit-breakers, when used with compound generators, should be connected in the lead on the side opposite the equalizer. By this means the total current from the machine passes through these instruments, whereas if placed in the other lead they would not be influenced by the current that goes by way of the equalizer.

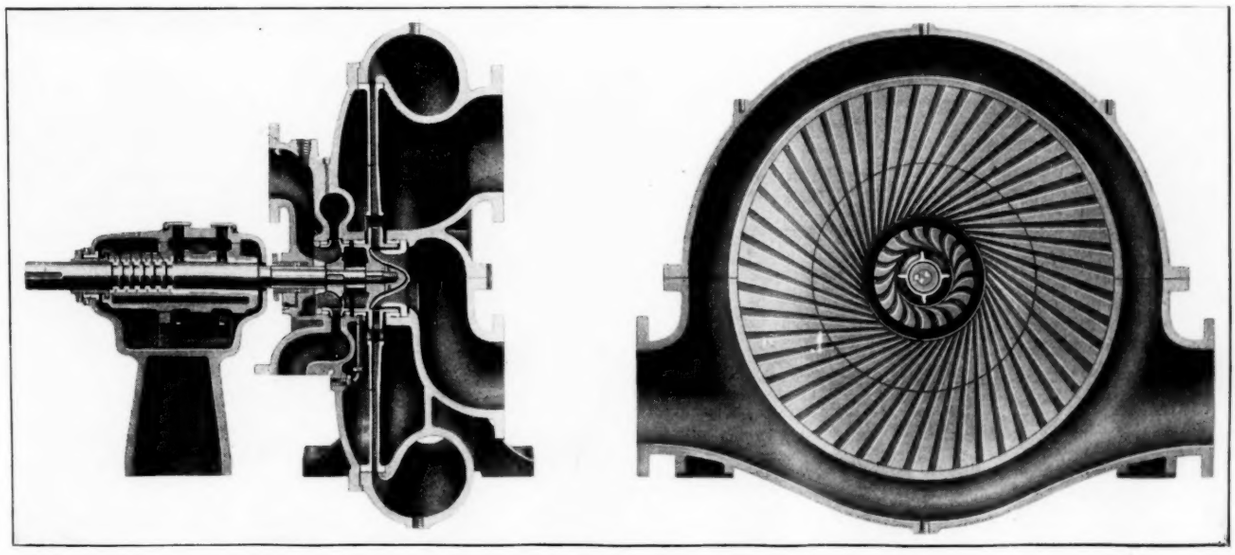
When two generators of like capacity are to be paralleled, the need for an equalizing connection may be avoided by "mutual excitation," in which the armature current from one generator is passed through the series field of the other. The connections are shown in Fig. 4. Should the voltage of one of the machines increase, it would take more than its share of load, and this increased current operating through the series fields of the second machine would raise its voltage, while the lesser load of the second, operating through the series field of the first, would lower the voltage of that machine. There would thus be a mutual action between the two, tending to an equal division of the load. It will be observed, however, that this scheme of connection is only possible when both generators are run at the same time, unless some system of switches is used whereby each machine is excited by its own series field when running alone.

While it is not usual to run series generators in parallel on constant potential circuits, it is interesting to note that their case is analogous to that of compound generators in parallel and that all that has been said with respect to the latter applies equally to series machines. If connected in parallel without an equalizer, one machine would be most certain to overpower and reverse the other, and this would be even more likely to occur than in the case of compound generators, on account of there not being a shunt field to oppose the reversing action of the series field. With an equalizing connection, however, there would be no difficulty in paralleling two or more series generators. Multiple excitation is also possible, as in the case of compound generators, and allows the equalizer to be dispensed with.

Alberger Centrifugal Air Pump

A new design of hydraulic centrifugal dry-air pump has been recently placed on the market. The sectional views show the design in detail.

The principle of operation is the use of intermittent pistons of water to expel the air entering the pump from either a surface, jet or barometric condenser and thus produce a vacuum. A high-speed inclosed impeller hurls water in separate streams across an air gap into a compression, or diffusion, vane chamber, in which the vanes cut the water into a large number of layers forming



SIDE AND END SECTIONAL VIEWS OF THE ALBERGER CENTRIFUGAL AIR PUMP

pistons separated by the entrapped air, which is thereby forced to the outer edge of the diffusion chamber, through the discharge and into the sump, from which the air is cleared.

The casing of the pump is of cast iron and with the side plates is divided on the horizontal center line, permitting convenient removal for inspection, the only openings in the upper half of the casing being arranged with vertical flanges so that pipe connections can be easily broken in case it is necessary for any reason.

The hurling-water inlet is in the lower half of the casing from which the water passes to the bronze impeller, provided with water-sealed sealing rings, thus avoiding air leakage. The diffuser, or compression chamber, with its vanes, is bronze, the tips of the vanes being made separate to permit inexpensive replacement after long-time service.

A single heavy water-cooled pedestal bearing carries the shaft, and incorporated in this bearing is a multi-collar marine-type thrust with an adjusting device to maintain the rotating element in correct lateral position under the conditions met with in practice.

This pump is manufactured by the Alberger Pump and Condenser Co., 140 Cedar St., New York City, either with or without a condensation removal pump, as shown in the sectional view, for high-vacuum.

Selection of Stacks and Breeching

By William Wippermann*

SYNOPSIS—A convenient chart by which a brick or steel stack or breeching may be properly proportioned. Temperature and volume of flue gases, temperature of outdoor air, horsepower served and draft loss, all enter into the final results. Specific examples are solved to show the working of the chart.

The determination of the height and diameter of a chimney from tables or formulas connecting height and diameter with horsepower often results in an impossible stack. As a rule there are so many factors involved that no one table or formula meets the requirements of the individual problems. But it is possible to present the data on a chart in such a way that chimney and breeching dimensions can be determined with ease and certainty. That was the aim in designing the accompanying chart.

The function of a chimney is to carry off the gases and to produce draft. Its height is fixed by the draft necessary to overcome the resistance of bends, damper, connections, breeching, boiler and grate. This resistance is not a function of the horsepower, but varies with every installation. The cross-sectional area must be so large that the stack will handle the volume of gas without excessive loss in friction. By increasing the height, the diameter can be reduced, and conversely, but among the various chimneys that are large enough, there is one that is cheaper to build.

The draft available for producing flow is usually measured in inches of water column. Its intensity depends upon the difference in weight of the outside and inside air columns, reduced under flow conditions by the friction loss in the stack. For a given temperature and pressure it is a maximum when no gases are flowing. The weight of a cubic foot of gas depends upon the temperature, pressure and nature of the gas. For chimney calculations flue gas is always considered to be the same as air. Inasmuch as draft is always measured in inches of water column, the scale on the chart shows the weight of air in inches of water column.

The use of this chart will be illustrated by determining the dimensions of a chimney and breeching for a boiler house containing five 350-hp. water-tube boilers vertically baffled and equipped with chain grates burning Illinois, or equal, coal. The overload is not to exceed 50 per cent. of rating. The stack temperature will average 520 deg. F. at maximum rating and about 440 deg. F. at normal rating. The outdoor temperature will be taken at 60 deg. F. and the atmospheric pressure at 14.5 lb. per sq.in.

Referring to the chart, the intersection of a vertical representing 520 deg. F. with a diagonal representing 14.5 lb. per sq.in. determines a horizontal through 0.0077 in., the height of a water column 1 ft. square that is of the same weight as a cubic foot of air at the given pressure and temperature. The same method shows that 0.0145 in. of water weighs the same as a cubic foot of air at a temperature of 60 deg. F. and a pressure of 14.5 lb. per sq.in. The difference between 0.0145 and 0.0077 is 0.0068 in. This is the potential draft per foot of stack when no gases are flowing. Under conditions of flow it must be reduced by an amount equal to the friction per foot of stack.

The head lost in friction is a function of the square of the velocity. It depends upon the roughness of the surface, on the density of the gas and on the surface in contact with the gas. For a brick stack,

$$D = 54\frac{W^2TH}{d^5P}$$

in which

D = Draft loss in inches of water column;
W = Pounds of flue gas flowing per second;
T = Absolute temperature, degrees F.;
P = Absolute pressure, pounds per square inch;
H = Height in feet above center of breeching;
d = Diameter in inches;
54 = Factor depending on the roughness of a brick stack.

The atmospheric pressure is fixed by the altitude. The temperature of the flue gas depends upon the heat-absorbing qualities of the equipment. The quantity of flue gas W depends upon the quantity and quality of coal burned. It is represented on the chart by diagonal lines under the caption "Pounds of Flue Gas per Boiler Hp.-Hr.," which can be estimated from the coal and CO_2 guarantees made by the stoker and boiler manufacturers, or it can be estimated from experience. It varies from 70 to 130 lb. per hp.-hr. In this problem it will be taken as 110.

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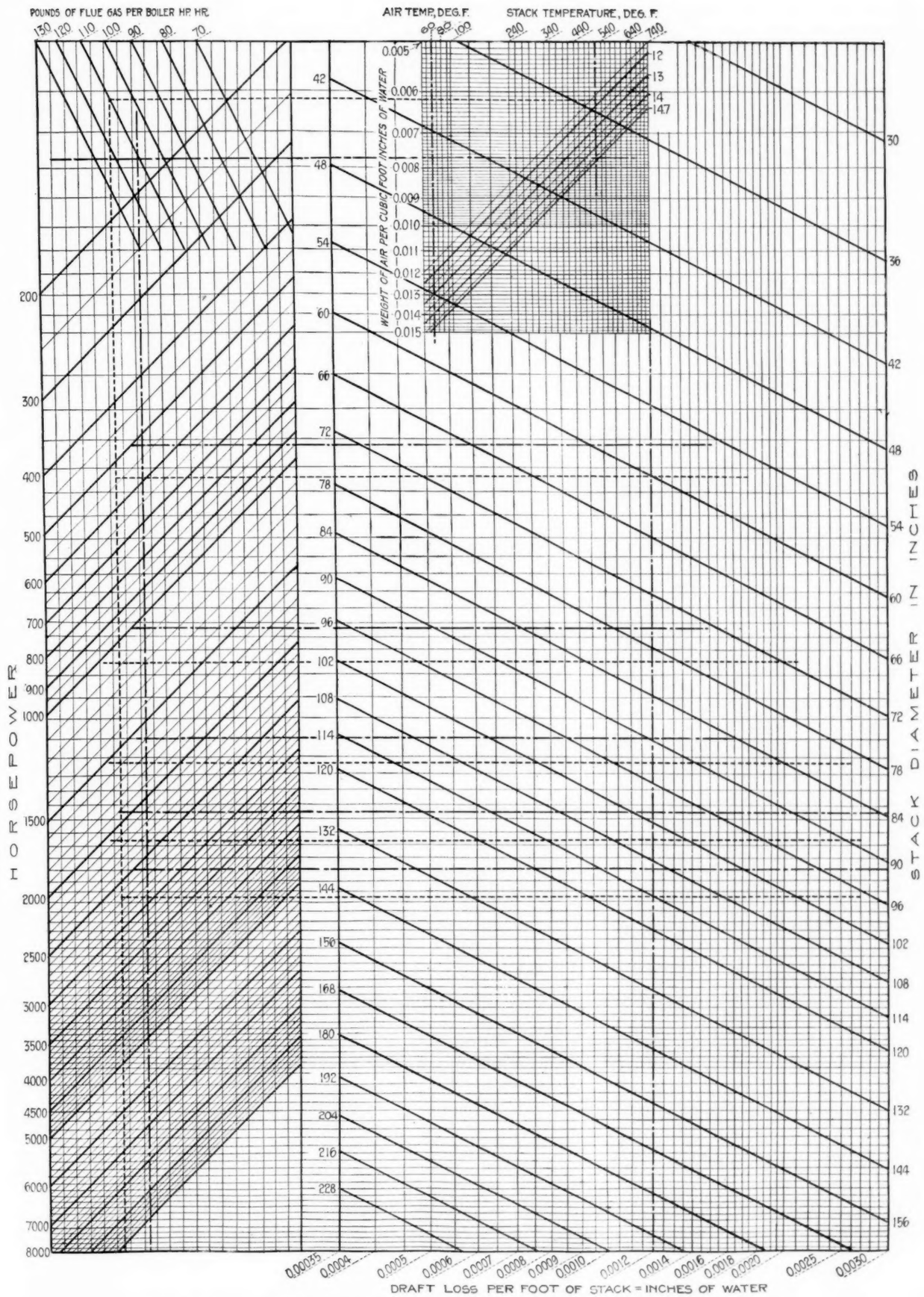

CHART FOR DETERMINING THE DIMENSIONS OF CHIMNEYS AND BREECHINGS

The chart is a graphical solution of the formula previously given. The draft loss per foot of stack D is determined as follows: Through the intersection of the vertical through 520 deg. F. and the diagonal 14.5 lb. per sq.in. draw a horizontal line intersecting the diagonal marked 110 lb. of flue gas per hp.-hr. Through this intersection draw a vertical cutting all of the horsepower diagonals. Inasmuch as this line is referred to several times, it will be called the *"T, P and W"* line. Continuing, draw a horizontal line through the intersection of the *"T, P and W"* line with the horsepower line 2,625 (150 per cent. of the rating of the five boilers). This line intersects some of the stack diameters, and the reading of the intersection on the "Draft Loss per Foot of Stack" scale is D in "Inches of Water" column. The values of D for different diameters are tabulated here:

Stack Diameter, In.	Draft Loss	Stack Diameter, In.	Draft Loss
132	0.0005	108	0.0013
120	0.00075	102	0.0017
114	0.0010	96	0.0023

Under conditions of maximum flow the draft available for producing flow is the difference between the potential draft and the friction loss. If the total draft required is known, the height of stack necessary can be computed by dividing the draft required by the available draft per foot of stack.

Before the height of the stack can be computed, an estimate must be made of the drop through the furnace, boiler, breeching, etc. The furnace draft, the drop through the boiler and the drop through the economizer, if one is used, can be furnished by the manufacturers. It is best to get these data from them because there is considerable variation in the different types of apparatus. It is customary to figure on a drop of 0.1 in. per 100 ft. of breeching of the proper size, on a loss of 0.05 in. for every right-angled bend; 0.05 in. for every boiler connection or entrance to breeching and breeching entrance to stack and 0.05 in. for a damper of proper design—that is, one that offers resistance at the lower capacities and a minimum of resistance at high ratings when maximum draft is required. Additional obstructions must be allowed for.

For the given problem the furnace draft at 150 per cent. of rating is 0.40 in., the drop through the boiler and damper is 0.47 in., the drop in 80 ft. of breeching is 0.08 in.; and the draft loss in entrance is 0.30 in., making a total draft required at the stack of 1.25 in. The heights of stack that will produce this draft are given in the following table. As previously figured, the potential draft per foot of stack is 0.0068 in. Deducting from this the draft loss per foot of stack, as itemized in the foregoing table, gives the available draft per foot of stack. Dividing the total draft required by the draft per foot of stock gives the height of stack.

Stack Diameter, In.	Available Draft, In.	Stack Height, Ft.	Stack Diameter, In.	Available Draft, In.	Stack Height, Ft.
132	0.0063	198	108	0.0055	228
120	0.0060	208	102	0.0051	245
114	0.0058	216	96	0.0045	278

The total height of the stack is the sum of the foregoing and the distance from the center of the breeching to the base. In the second paragraph it was stated that the cross-section of a stack must be large enough to handle the gas without excessive friction loss, and by increasing the diameter and conversely a size of stack could be found that was cheapest. With the exception of the last two stacks all are equally good and well-proportioned. One of them, however, is cheaper than the rest.

But for the following exception, the method of figuring the sizes of steel stacks is the same as for brick. The draft lost in friction in a steel stack is less than that lost in a brick stack because the resistance is less. It is 0.08 of the draft lost in brick or brick-lined chimneys. For a steel stack for the boiler house in question the friction head is tabulated here:

Stack Diameter, In.	Friction Head, In.	Stack Diameter, In.	Friction Head, In.
132	0.0005×0.8 = 0.0004	108	0.0013×0.8 = 0.0010
120	0.00075×0.8 = 0.0006	102	0.0017×0.8 = 0.0014
114	0.0010×0.8 = 0.0008	96	0.0023×0.8 = 0.0018

Quite frequently it is necessary to estimate the capacity of a stack already built. Take for example a stack 120 in. diameter and 208 ft. high. Using the data of the previous problem, the draft required at the entrance to the stack is 1.25 in. of water column. The available draft per foot of stack is $1.25 \div 208 = 0.0060$. The difference between the potential draft (0.0068 in.) and the available draft (0.0060) is 0.0008, the loss per foot of stack. The intersection of the 120-in. diameter line with a vertical through 0.0008 in. locates a horizontal line. The intersection of this horizontal with the *"T, P and W"* line fixes the capacity of the stack. It happens to be 2,625 hp. in this problem.

In tall buildings the height of the stack is fixed by the height of the building. If the boiler house in question is in the basement of a tall building such that the stack rises 300 ft. above the breeching connection, the solution for the diameter is as follows:

Estimating the Draft

The estimated draft required at the stack is 1.25 in. The available draft per foot of stack is $1.25 \div 300 = 0.0042$. The difference between the potential and available draft is $0.0068 - 0.0042 = 0.0026$, the friction head in inches of water. The intersection of a vertical through 0.0026 in. and a horizontal through 2,625 hp. fixes the diameter. In this case the nearest diameter is 96 in.

In making up the estimate of the draft drop in the breeching, the figure 0.1 in. per 100 ft. of flue was used. This is safe for round steel flues of the proper size, but for square, rectangular and brick flues the retarding effect is greater, consequently the area of such flues should be larger than that of a round steel flue. It is not necessary that the breeching be of the same area throughout. It is best to proportion the cross-section according to the volume of gases passing.

The chart, which was designed for brick flues and stacks, can be used to advantage in computing the cross-section of the breeching. As mentioned before, the draft lost in a steel flue is 0.8 in. of that lost in a brick flue. Consequently the figure 0.0012 in. on the "Draft Loss per Foot of Stack" scale represents 0.0010 in. for steel flues, because 0.0012×0.8 is practically 0.0010 in., the draft drop per foot of flue used in making up the total draft estimate.

The cross-sections are determined in the following manner: Through 0.0012 in. draw a vertical line and through the intersection of the horsepower diagonals 525, 1,050, 1,575, 2,100 and 2,625 with the *"T, P and W,"* line draw horizontal lines. The intersection of the 0.0012 in. vertical with the horizontal lines fixes the diameter of the flue. Inasmuch as a round breeching is seldom used, it is better to make it square than rectangular, making the side of the square equal to the diameter determined

from the chart. Then the cross-section will be a trifle larger than necessary. The dimensions appear in the following table:

Horsepower	Cross-Section
525	4 ft. 9 in. square
1,050	6 ft. 4 in. square
1,575	7 ft. 5 in. square
2,100	8 ft. 4 in. square
2,625	9 ft. 2 in. square

If rectangular cross-sections are used, the area must be increased because a rectangular section offers more resistance. General practice is to make them up to 120 per cent. of the area of the stack. If the flue is lined with tile or is of brick, the vertical line is drawn through 0.0010 in. This results in proportionately larger flues, because the resistance is greater.

Method of Chimney Analysis

This method of chimney analysis is simple and easily learned. It cannot very well result in impossible stacks. Many of the chimney men use velocities in figuring areas. A proper analysis shows that high velocities are used in large-diameter stacks and low velocities in the small-diameter stacks. The velocity may vary from 15 to 40 ft. per sec., depending on the size of the stack. Judgment of course is necessary in estimating the temperature of the flue gases, pounds of flue gas per boiler horsepower and the draft required at the stack. In making up the item last named, due consideration must be given to the fact that more draft is required for higher altitudes than for the sea level. The items to be considered at the higher altitudes are the drop through the boiler and the furnace draft, and they can be estimated with ease by remembering that altitude has the same effect as increasing the capacity—more draft is needed. The cross-section of the breeching and the increased height necessary to overcome the increased friction loss in the stack are taken care of in the method of computing the respective dimensions.

If the atmospheric pressure had been 11.7 lb. per sq.in. instead of 14.5, the furnace and boiler drop for the specific problem under consideration would increase from 0.87 to 0.97 in., making a total of 1.35 in. of draft required at the stack. The solution of the problem may be traced by the light dotted lines on the chart. The results obtained follow:

CHIMNEY ANALYSIS

Potential draft, in. of water column	0.0117—0.0062=0.0055
Draft loss (120 in. diam.), in. of water column	0.00095
Draft loss (132 in. diam.), in. of water column	0.0006
Available draft (120 in. diam.), in. of water column	0.0045
Available draft (132 in. diam.), in. of water column	0.0049
Height of 120-in. stack above breeching, ft.	300
Height of 132-in. stack above breeching, ft.	275

BREECHING ANALYSIS

Horsepower (150 per cent. rating)	Cross-Section
525	5 ft. 0 in. square
1,050	6 ft. 8 in. square
1,575	7 ft. 9 in. square
2,100	8 ft. 9 in. square
2,625	9 ft. 6 in. square

These results check very well with dimensions obtained by using the well-known methods of correcting for altitude. Chimney and breeching dimensions can be figured with ease and sufficient accuracy by drawing a few lines on charts ruled as shown. Such a chart is a permanent record of the analysis of the chimney and breeching.

A Filter will soon recover its cost in oil reclaimed. Using oil repeatedly without thoroughly cleansing it is likely to cause trouble.

Frictional Resistance of a Bearing is the resisting torque, or so-called moment of friction, or it may be defined as the quotient of the resisting torque by the mean radius of the journal.

Ford Check and Stop Valves

The Ford automatic check and stop valve, Fig. 1, is designed without the usual dashpot, to prevent chattering of the valve disk, which is unbalanced. The disk has an apron *A*, so that by the time the steam is flowing from the boiler into the main in any appreciable quantity, the disk is enough above its seat to prevent wire-drawing.

A feature of the valve is the adjustable outside spring *B*, which permits of a flow of steam from the boiler to the main at an even pressure or at a boiler pressure any given small excess over the existing pipe-line pressure

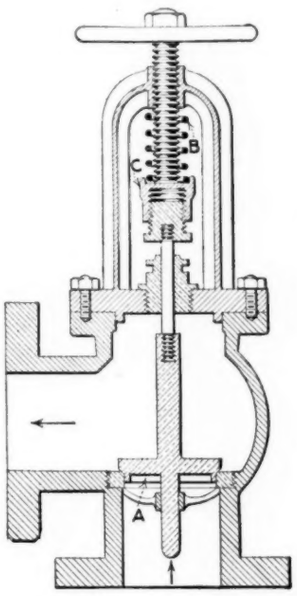

FIG. 1. FORD CHECK AND STOP VALVE

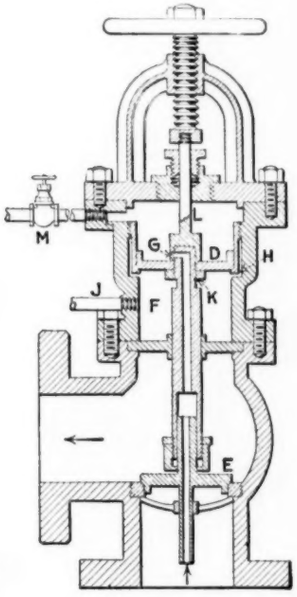

FIG. 2. FORD CHECK NONRETURN VALVE

that may be desired, by adjusting the attachment *C*. The valve spindle running through the packing gland gives visible proof of the valve operation.

The valve shown in Fig. 2 acts as a check valve against flow of steam back into the boiler from the main, as an automatic emergency valve to isolate the steam main from a boiler or boilers carrying normal pressure and as a hand-operated stop valve.

The emergency piston is a loose fit and during normal periods is raised from its seat *H* and is also disengaged from the check valve *E*, which allows it to operate automatically. The chamber *F* is connected to the steam main by the pipe *J*, and if the pressure in the chamber drops, the boiler pressure acts on top of the piston *D*, coming through the hole *G* in the valve stem, and causes the piston to travel downward, engaging with the check valve *E* to close it. When the piston *D* reaches the seat *H*, a tight piston is insured without the use of rings or packing. By a screw thread the piston may be adjusted on the spindle, and locked in position by the nut *K*. The traveling rod *L* runs through the packing gland and is a means of indicating what the valve is doing.

After an emergency shutdown the valve can be put in service by opening the starting valve *M* that is placed in the front of each boiler, convenient for manipulation.

This valve is manufactured by the Thomas P. Ford Co., Inc., 410 Broome St., New York City.

How Much Will We Save?--II

By Carleton W. Hubbard*

SYNOPSIS—Snyder demonstrates to Young that a pump is cheaper to operate than an ejector, and gives figures to prove it.

"Hello, Snyder, this is John Young, at Easton & Weston's. Do you remember that ejector I was talking about to you the other day? Well, I've the measurements and dope you told me to get, and I'd like to know how much we'd save by putting in a pump in place of it. Have you got time to help me out?"

"Sure thing; come on down this afternoon."

In Easton & Weston's there was, near the engine room, a tank into which the cooling water from the condenser emptied. This water was then pumped to another tank about twenty feet higher by an ejector made up of ordinary pipe fittings. The second tank had coils of pipe in it, through which the exhaust from a small dynamo engine passed. From the second tank the water flowed to the water softener and thence to the boilers.

The sketch shows the arrangement. Young had made all the measurements of the differences in elevation and had been watching the temperature of the water in Tank No. 1 and at the outlet of the pipe by which the water entered tank No. 2. He had found that when the water in No. 1 was 146 deg., that entering No. 2 was 178 deg. When he dropped in on Snyder that afternoon, they got busy at once.

Determining the Water Pumped

"The first thing, Young, is to find out how much water you are now pumping with one pound of steam. The water goes into that ejector at 146 and comes out at 178 deg.; that's a rise of 32 deg. That means that each pound of water absorbs 32 B.t.u. in going through the ejector. Now it gets that heat from the steam, so we've got to know how much heat there is in one pound of steam. You said that the steam pipe for that ejector is taken from a separator drip, so it's probably pretty wet. I wouldn't be surprised if it had 8 per cent. of moisture. That brings up the question of how the heat in one pound of steam is divided. Let's see, you carry 100 lb."

Snyder reached for his steam tables and ran his finger down the column headed "pressure" until he reached the point corresponding to 100 lb. gage, or 115 lb. absolute. "The next column gives the temperature of the steam, 338.1 deg. (Marks and Davis) and the fifth column the total heat in one pound, 1,188.8 B.t.u. This total heat is divided into two parts—the 'heat of the liquid,' which is 309 B.t.u., and the 'latent heat of evaporation,' which is 879.8 B.t.u. The sum of these is the total heat."

"You're getting in pretty deep water for me, Snyder. Can't you explain that a little more?"

"Sure; look at it this way. For every pressure water has a certain boiling point. You've heard that water boils at 212 deg. It does—when the pressure is atmospheric or 14.7 lb. absolute, which would be the same as zero on your gage. Now the higher the pressure the higher the boiling point. Here, with a gage pressure of 100 lb., or 115 lb. absolute, using the nearest figures given in the table, the boiling point is 338.1 deg. The 'heat of the liquid' is simply the amount of heat required to raise one pound of water from the freezing point to its boiling point, which depends on the pressure. Now after you've got the water to the boiling point, you've got to add more heat to turn it into steam. This is the 'heat of vaporization.' If you add these two quantities of heat together, you get the total heat in one pound of steam. Suppose we take a pound of water at freezing temperature and decide that we will turn it into steam at 100 lb. gage. First we've got to add the 'heat of the liquid' to the water—that amounts to 309 B.t.u., as the tables show. Then we've got to add the 'heat of vaporization' to turn that pound of water into steam."

"That clears it up some. I've had some steam tables for a long time, but I never could seem to find out how to use them. When you're told how to do a thing it seems a lot easier than when you read how to do it."

"You'll soon get the hang of it. Now there is another point to consider here. A pound of the stuff you take from that separator drip isn't all steam. We don't know how much of it is water, but we estimate that 8 per cent. of it is. That leaves 92 per cent. steam. To get the total heat in one pound of a mixture like this, you take the 'heat of the liquid' for one pound, 309 B.t.u., for all the water has been raised up to the boiling point. To that you must add a part of the 'heat of vaporization,' depending on how much of the mixture is steam. In this case, where the moisture is 8 per cent., only 92 per cent. of a pound is in the form of steam, so to the 'heat of the liquid' you want to add only 92 per cent. of the 'heat of vaporization.' So you multiply 879.8 by 0.92, which equals 809.4, and add 309. The sum of these, 1,118.4 is the number of B.t.u. in one pound of the mixture of steam and water which you supply to the ejector.

B.t.u. in 1 Lb. of Water

"You'd get that much heat if you cooled that mixture down to the freezing point, but you cool it down only to the temperature of the water discharged from the ejector, and that is 178 deg. That's 146 deg. above the freezing point, and as it takes 1 B.t.u. to raise a pound of water 1 deg., there are in that mixture 146 B.t.u. that you can't get. So if you subtract 146 from 1,118.4, you will get 972.4 B.t.u., which is all the heat available in one pound of the mixture you take from your separator to heat the water passing through your ejector. The heat supplied to the water is equal to that given up by the steam. Now we saw that one pound of water absorbed 32 B.t.u. in passing through the ejector, and we have just figured that one pound of the mixture of steam and water supplied could give up 972.4 B.t.u. So if you divide 972.4 by 32, you will get the number of pounds of water pumped by one pound of steam. That equals 30.4. "The next thing is to find how far this water has to be lifted. The distance from the engine-room floor to the water level in tank No. 1 is 7 ft. 8 in. + 4 ft. 4 in., or 12 ft. From the floor to the top of the pipe leading into tank No. 2, the distance is 17 ft. 3 in. + 13 ft. 9 in., or 31 ft. The height

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through which the water must be pumped is 31 — 12 = 19 ft. Now we must get an idea as to how much water you use in a year. You haven't a water meter, have you?"

"No."

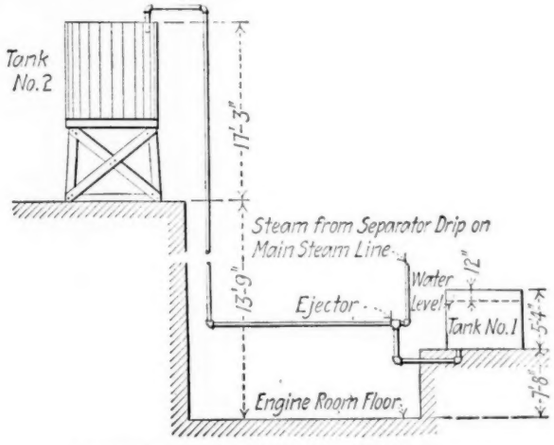
"Do you know how much coal you burn?"

"Yes, about 7,500 tons a year."

"That's about twenty tons a day 365 days in the year. Do you have your coal analyzed?"

"No. I tried to get them to have it done, but they couldn't see where they would save anything."

"Well, we'll have to do a little guessing then. If you're getting an average semibituminous coal and are working



ARRANGEMENT OF TANKS AND PIPING

at 65 per cent efficiency, you ought to be evaporating about 8.5 lb. of water per pound of coal. Twenty tons a day, 2,240 lb. to the ton, 8.5 lb. of water per pound of coal: that's 380,800 lb. of water a day. Now we figured that one pound of steam raised 30.4 lb. of water from tank No. 1 to tank No. 2, so if you have to pump 380,800 lb. of water a day you have to use $\frac{380,800}{30.4} = 12,526$ lb. of steam per day."

"Gosh, Snyder, that's a bunch of steam."

"It is, Young, but let's see what a pump would take. First we'll find out how many pounds of water must be pumped each minute to give 380,800 lb. a day. This will be $\frac{380,800}{24 \times 60} = 265$ lb. per min. This has to be raised 19 ft., and the horsepower required will be $\frac{265 \times 19}{33,000} = 0.1525$ hp. That would be the horsepower required on the water end of the pump. If your pump had an efficiency of 50 per cent., that would mean that the steam end would have to develop twice as much horsepower, or 0.305 hp. We'll play it safe and assume that the pump you put in will be a poor second-hand one and will require 200 lb. of steam per horsepower per hour. Then the steam required per hour would be 200 × 0.305 = 61 lb., or 61 × 24 = 1,464 lb. per day. Under the present method you use 12,526 pounds per day, so that by making the change you will save 12,526 — 1,464 = 11,062 lb. of steam per day."

"That's a whole lot, Snyder, but you haven't gone far enough to suit me. You told me once that every power-plant waste goes back to the coal pile. They pay money for coal, but they can't seem to see that steam costs them anything. How do we get back to the coal pile and get the saving into dollars."

"That's easy, John. If you evaporate 8.5 lb. of water per pound of coal and you save 11,062 lb. of steam, your saving in coal will be $\frac{11,062}{8.5} = 1,301$ lb. a day. For a year that would amount to $\frac{1,301 \times 365}{2,240} = 212$ tons, nearly. You are paying $3 a ton, so the money saving would be $636. The pump installed ought not to cost over $200, so it would certainly be a good proposition."

"You're right, Snyder, and I'm much obliged to you."

"That's all right, John. Drop in again when you have another problem, and we'll figure it out together."

Action of Air in Surface Condensers

By C. F. Hirshfeld*

SYNOPSIS—A study of some of the phenomena connected with the presence of air in a surface condenser and the effects shown graphically. The author shows that much of the air leaks into the condenser and that very little follows through from the boiler. Pressure and temperature relations, variation in volume of the air and steam and the results from varying the air leakage into a condenser in operation are given attention.

The function of the surface condenser and its auxiliaries in a modern plant is twofold: First, it maintains a low pressure in the space into which a prime mover exhausts, and, second, it liquefies practically all of the steam exhausted from the prime mover, so that the resulting pure and warm water may be returned to the boilers through heaters as feed water. The advantages resulting from low back pressure and pure and warm feed water have been exhaustively discussed and are, in general, well understood by all plant operatives. There seems, however, to be more or less mystery connected with the process going on in the condenser, particularly with regard to the action of air and the effect which it has upon the vacuum attained.

One of the most important questions to be answered in this connection is, Where does the air that must be continuously withdrawn from surface condensers come from? Some of it enters the prime mover with the steam, having entered the boiler in solution in the feed water. This air is carried through the steam lines and the prime mover with the steam and ultimately enters the steam space of the condenser. With a properly operated surface-condensing plant, the quantity of air entering the condenser from this source is small. By far the larger portion of the air enters through leaks in

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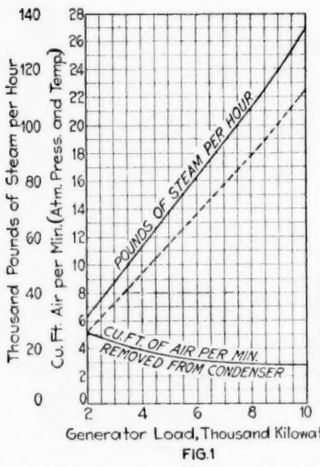
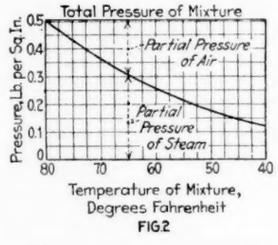
those parts of the apparatus which are operated with internal pressures below that of the atmosphere.

This can be appreciated best after an inspection of Fig. 1. The curve of air pumped out of the condenser was actually obtained by measuring the air drawn from a condenser serving an 8,000-kw. turbine. The curve of total steam used by this turbine is given to show the variation of this quantity with the load. Had all of the air originally come from the feed water, the quantity of air would have increased in the same proportion as did


Thousand Pounds of Steam per Hour
Cu. Ft. Air per Min. (Atm. Press. and Temp.)
POUNDS OF STEAM PER HOUR
CU. FT. OF AIR PER MIN. REMOVED FROM CONDENSER
Generator Load, Thousand Kilowatts
FIG. 1


Total Pressure of Mixture
Partial Pressure of Air
Partial Pressure of Steam
Pressure, Lb. per Sq. In.
Temperature of Mixture, Degrees Fahrenheit
FIG. 2

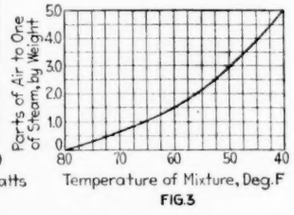


FIG. 3

FIGS. 1 TO 3. CURVES OF CONDENSER PERFORMANCE

Fig. 1—Air pumped from a condenser serving an 8,000-kw. turbine. Fig. 2—Component pressures of air and steam at different temperatures. Fig. 3—Ratios of air and steam at different temperatures.

the weight of steam used, as shown by the dotted line in the figure. Obviously the air must have entered largely through leakage and, moreover, the amount of leakage decreased greatly as the load increased. This is characteristic of all such installations. Assuming that leakage was reduced to zero at the maximum load, all of the air shown for that load must have come from the feed water. This being true there would have been 0.000095 lb. of air per pound of feed water, which is obviously small. In reality leakage could not have been entirely eliminated, and the amount of air in the boiler feed must have been still less than this figure would indicate.

Since most of the air enters through leakage and since, as will be shown later, the presence of air in the condenser is decidedly detrimental, it is important to keep the apparatus as air-tight as possible. It may be mentioned in passing that the performance shown by the curve of Fig. 1 is representative of good practice.

The effect of this air on the condenser and its auxiliaries may now be investigated. For this purpose it will be necessary to know just how a condenser and its allied apparatus operate. The back pressure against which a prime mover exhausts is lowered so that the working substance may expand to a pressure below atmospheric and deliver a proportionately greater quantity of energy. Theoretically this lower pressure, or vacuum, could be maintained by pumping the working substance out of a vessel as fast as the prime mover exhausted it into that vessel. The pump would have to raise the pressure from that in the vessel to atmospheric in order to be able to discharge the working substance, and theoretically, this would require just as much energy as had been liberated by expansion from atmospheric pressure down to the pressure maintained in the vacuum. Practically, it would require a great deal more, so that the process would result in a loss and a pump as large or larger than the prime mover would have to be used.

If it were possible to reduce the volume of the working substance after it had been exhausted from the prime mover and before it entered the pump, the pumping work would be decreased and the size of the pump would be reduced. If the working substance were all steam, this would be simple, as its volume could be reduced until negligible by liquefying, or condensing, it before it entered the pump to have its pressure raised sufficiently high to enable it to pass out against atmospheric pressure. It is such a process that is approximated in a real condenser, but it is greatly modified by the presence of air that is obviously noncondensable under the conditions existing in such apparatus and that must therefore be removed from the condenser by a pump that is capable of raising its pressure from that maintained in the condenser to a pressure approximately equal to that of the atmosphere.

The function of the condenser is to remove, by condensation, as much as possible of the steam that enters the apparatus mixed with the air, so that so far as possible the air pump will be required to pump air only.

When two gaseous materials such as steam and air occupy the same vessel at the same time, each behaves as though the other were not present. Thus each will exert upon the walls of that vessel the same pressure as it would if it alone occupied the vessel. The total pressure on the walls will therefore be the sum of the pressures exerted by the two materials respectively. This action is pictured in Fig. 4. The two springs shown both occupy the space within the cylinder, and both rest on the bottom of that cylinder. Each spring can be compressed any amount desired and can thus be made to exert any pressure, within limits, upon the bottom of the cylinder. The total pressure on the bottom of the cylinder will always be the sum of the pressures exerted by the two springs respectively.

FIG. 4

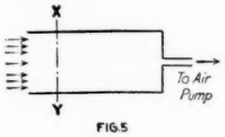
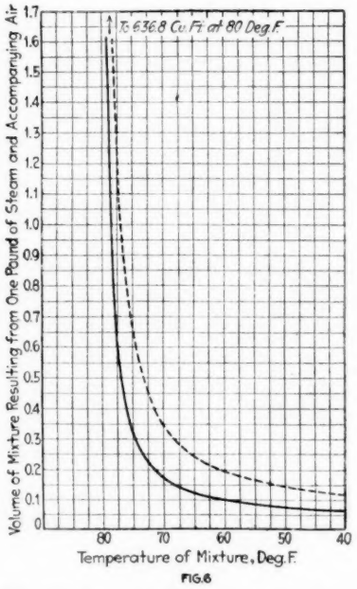


FIG. 5


To 636.8 Cu. Ft. at 80 Deg. F.
Volume of Mixture Resulting from One Pound of Steam and Accompanying Air
Temperature of Mixture, Deg. F.
FIG. 6

FIGS. 4 TO 6. DETAILS OF CONDENSER PRACTICE

Fig. 4—Illustrating air and steam pressure. Fig. 5—Diagrammatic sketch of condenser. Fig. 6—Volume of mixture at different temperatures.

The pressure at any point within a condenser must similarly be equal to the sum of the pressures exerted by the water vapor or steam and by the air present at that point. But under the conditions existing in a condenser, the steam is always saturated and the pressure which it exerts is definitely connected with its temperature, as shown by the steam tables. Thus if the temperature of the mixture happens to be 80 deg. F., the pressure exerted by the steam will be 0.505 lb. abs. per sq.in.; if the temperature of the mixture happens to be 60 deg. F., the pressure exerted by the steam will be 0.2562 lb. abs. per sq.in.

These facts may now be applied to the study of an ideal condenser. Assume for this purpose a vessel such as shown in Fig. 3, with a steam-air mixture entering continuously at the left end and so arranged that steam can be condensed out of the mixture as it flows toward the right. That part of the mixture that leaves through the small pipe at the right may be assumed to be received by an air pump and to be delivered to the atmosphere. Suppose the mixture enters at 80 deg. F. and that it contains 0.000126 lb. of air per pound of steam, a proportion corresponding approximately to the conditions shown in Fig. 1 for a load of 8,000 kw. The pressure exerted by the steam at 80 deg. F. is given in the steam table as 0.505 lb. abs. The pressure exerted by 0.000126 lb. of air at 80 deg. F. and occupying the same volume as a pound of steam at that temperature can be determined from the laws of gases. The value is about 0.00004 lb. per sq.in. The pressure of the mixture at entrance must then be 0.505 + 0.00004 = 0.50504 lb. per sq.in.

Assume now that, by the time the mixture has reached the plane XY in Fig. 5, one-half of the steam has been condensed. This means that the weight of air originally mixed with one pound of steam must now be mixed with one-half pound of steam, or expressing it differently, air is present in the mixture at the plane XY in the proportion of 2×0.000126, or 0.000252, lb. per lb. of steam.

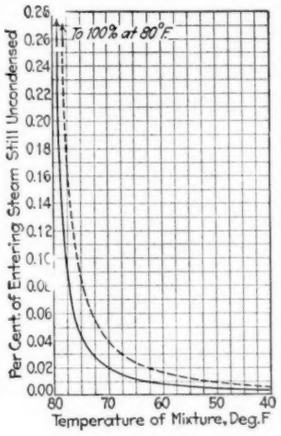


FIG. 7. STEAM UNCONDENSED AT DIFFERENT TEMPERATURES

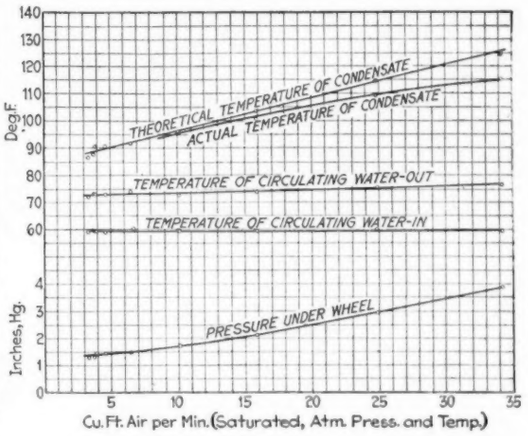


FIG. 8. RESULTS OBTAINED BY INCREASING AIR LEAKAGE INTO CONDENSER UNDER OPERATION

Pressure of Mixture in Condenser

In an ideal condenser the pressure of the mixture would be the same from end to end, and this will be assumed in this case. The pressure at the plane XY must therefore be 0.50504 lb. per sq.in. From the known properties of steam and air can be found the part of this pressure that is due to steam and the part that is due to air. Roughly, the pressure of the air will be about 0.00008 lb. per sq.in. and that of the steam about 0.50492 lb. The temperature of the steam will then be less than 80 deg. F., because at 80 deg. its pressure would be 0.505 lb. per sq.in. A similar line of reasoning applied to different points through the length of the condenser will show that as the steam is condensed the temperature of the mixture must be decreased if the pressure is to remain constant. Constant temperature would indicate a rise of pressure toward the discharge end of the condenser.

The effects of such a temperature change can be appreciated best after inspecting Fig. 2, which shows the partial pressures that steam and air must exert at different temperatures in the assumed case for the total pressure to remain constant at 0.50504 lb. per sq.in.

From the known air pressures at different temperatures and the known proportions of the material entering the condenser, it is possible to determine the proportions which the mixture must have at different temperatures. These values are shown graphically in Fig. 3. The rapid increase in the amount of air accompanying a given amount of steam as the temperature of the mixture is decreased is important.

Knowing the actual weight of air entering the condenser per pound of steam and having determined the ratio of air to steam at different temperatures, it is simple to determine next the volume of the mixture at different temperatures and the fraction of the steam that must have been condensed before each temperature was reached. The variation in volume is shown by the full-line curve in Fig. 6, and the percentage of steam still uncondensed at different temperatures is given similarly in Fig. 7.

Inspection of the volume curve in Fig. 6 shows that with decreasing temperature the decrease of volume is at first rapid, but that it afterward becomes less and less rapid. The smaller this volume the smaller the air pump required to pump the material out of the condenser. But it is difficult to lower the temperature of a mixture rich in air, so that if we attempt to reduce the temperature too much, the amount of surface required rapidly increases. In practice, therefore, a compromise must be made between the cost of pump and pumping and the cost of cooling surface. From this viewpoint the best results are generally obtained when the pump has a capacity approximately equal to that shown just above the knee of the curve, when that curve is drawn for average operating conditions.

Inspection of Fig. 7 shows that, to extract air at such temperatures under the condition here assumed, all but about 0.02 to 0.04 per cent. of the steam will have been condensed. It is also interesting to note that most of the steam is condensed at practically constant temperature. In the case assumed, all but about 0.25 per cent. had been condensed before the temperature dropped 1 deg. F.

The work of the condenser surfaces can thus be divided into two parts—that of condensing steam at practically constant temperature and that of reducing the temperature of the remainder with small condensation in order to

decrease the work required of the pump. As actually happens most of the surface is used for work of the first kind, only a small fraction of the surface being used for cooling.

We may now go a step farther and determine what would happen with increased air leakage. Suppose the capacity of the pump is limited to that required to remove the quantity of air previously assumed at a temperature of, say, 75 deg. F. If the quantity of air entering the air condenser is doubled, there must be a radical readjustment. Were we to assume the same initial steam temperature (80 deg. F.) and then to carry out all calculations as before, we should obtain volumes of mixture, as indicated by the dotted curve in Fig. 6, and percentages of steam uncondensed, as shown by the dotted curve in Fig. 7. The pressure then would be about 0.50508 lb. per sq.in. instead of 0.50504, as before.

If now some delivery temperature is chosen, say 75 deg. F. as before, the curves show that the volume of the mixture and the percentage of steam uncondensed are practically twice as great when the quantity of air per pound of steam entering is twice as great. This can be seen readily if it is remembered that the total pressure has been taken to be practically the same (0.50504 and 0.50508 lb. per sq.in.) and also the temperature at 75 deg. F. The partial pressure of the steam will be the same in both cases, because it is determined only by the temperature. Therefore the partial pressure of the air must be practically the same in both cases, as it is the difference between the total pressure and the part of that pressure which is due to the steam. If the pressure and the temperature of the air are the same in both cases, however, it follows that its volume must be twice as great in the second case, because there is twice as much present.

Displacement of Air Pump

Also, since the steam fills all the space occupied by the air, it follows that there must be twice as much steam uncondensed at 60 deg. F. in the second case, because there is twice as much space to fill. Obviously an air pump just capable of handling the mixture at 75 deg. F. in the first case would have about half the displacement required in the second case.

If this pump must be used in the second case, one of two things or a combination of the two must occur. Either the temperature of the mixture must be reduced to a much lower value, or the pressure of the mixture must be increased, or both of these things must occur. Any one of the three possibilities will decrease the volume.

Evidently from the volume curve, Fig. 6, to halve the volume possessed at 75 deg., the temperature would have to be reduced to below 70 deg. F., and this additional cooling might be impossible in the condenser as constructed or with water at the temperature available.

The only alternative would appear to be the raising of the pressure. At 75 deg. F. the pressure of the steam present is 0.4288 lb., and with a total pressure of 0.50508 the pressure of the air must be 0.50508 — 0.4288 = 0.07628 lb. To halve the volume the air pressure must be doubled, that is, raised to $0.07628 \times 2 = 0.15256$ lb., and the total pressure of the mixture would then be 0.4288 + 0.15256 = 0.58136 lb. per sq.in. But this pressure could not exist at the region in the condenser at which the mixture has a temperature of 75 deg. F. unless at least as high a pressure existed at the entrance to the condenser.

Therefore, unless the air is cooled to a lower temperature with increased leakage, the pressure in the condenser must rise, with double air leakage, from 0.50508 to 0.58136 lb. per sq.in. abs. in order that the air pump may be able to handle the air and noncondensable vapors. And this is approximately what happens in a real condenser. The pressure therein adjusts itself until the volume of the material at the temperature existing at the condenser air nozzle can be received and discharged by the pump. It is masked to a certain extent in real condensers by the drop in pressure that always occurs between inlet and outlet and by the fact that as the pressure throughout the condenser rises, the temperature head between steam and water changes, so that the performance of the cooling surface changes.

In general, however, the greater the air leakage the higher the pressure in the condenser, and therefore air leakage should be prevented as much as possible. This can probably be appreciated best from a study of Fig. 8, which shows results obtained with a real unit. Air was admitted to the unit at different rates while the load was maintained constant. The total amount of air handled by the air pump under each condition of leakage was determined by measuring the air-pump discharge with a large gasometer. The figure shows very well how sensitive the condenser pressure is with regard to air leakage.

De-emulsibility Testing of Engine Oils

By Frank T. Leilich

Oils for the lubrication of high-speed engines or other machinery employing a system of oiling in which the oil comes into contact with water should not emulsify. Many plants determine by trial the oil best suited to their conditions and confine their purchases to the proved brands. For a large consumer this plan is not desirable, as it eliminates competition. To permit this, but at the same time secure oils that will be satisfactory, it is customary to purchase the oils under specifications, which usually limit the composition, specific gravity, viscosity and flashpoint.

The results of tests on three engine oils offered under the same specifications are tabulated here:

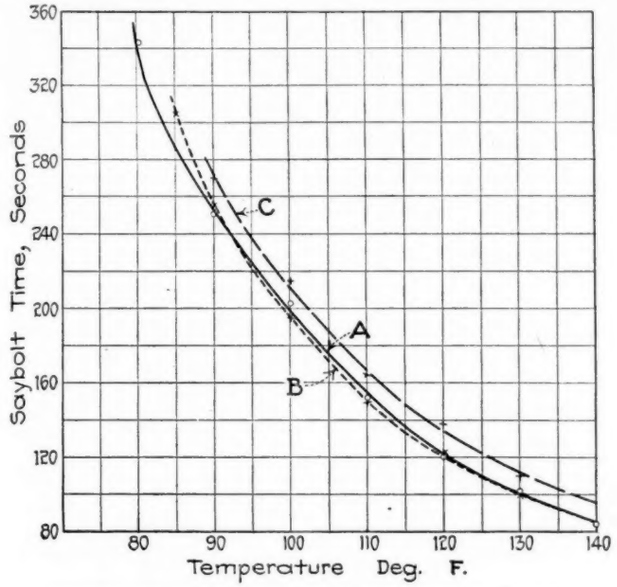
	A	B	C
Specific gravity at 60 deg. F.	0.8999	0.9088	0.8779
Flash, deg. F.	392	....	385
Free acid	None	None	None
Tars	None	None	None
Saponif. material	None	None	None

From an examination of the foregoing it might be inferred that in view of the almost identical character of the oils their lubricating qualities would be about the same. The viscosity curves of these oils are given in the illustration and for *B* and *C* are practically coincident. However, when used on the engines *A* was entirely satisfactory, *B* fairly so, but *C* totally unsuited. Thus it was clear that some other test would be necessary to determine the suitability of oils for actual service.

The question was taken up with the Bureau of Standards at Washington, and a test for deëmulsibility was suggested upon which the bureau has been working. This test consists of stirring for five minutes a mixture

of 20 c.c. of oil and 40 c.c. of water in a cylinder 1 in. diameter, the stirring being done by a flat paddle 7/8 in. wide by 3½ in. long rotated at 1,500 r.p.m. The deëmulsibility is determined by noting the rate of separation of the oil and water, this being expressed in cubic centimeters of separation per hour. The bureau, in the light of its present experience, recommends that the mixture be stirred and allowed to separate at a temperature of 122 deg. F.

Deëmulsibility tests of *A*, *B* and *C* gave the following results: *A*, 0+; *B*, 60; *C*, 94. These results bear out



VISCOSITY CURVES OF THREE SAMPLES OF OIL

the results of actual tests on the engines and lead to the belief that the deëmulsibility test, if incorporated into oil specifications, will furnish a reliable means of determining the suitability of a given oil.

Power from a Pumping Plant

By H. A. Megraw

It is usually considered that the power expended in operating a pumping plant is used without possibility of return, and any arrangement by which a part of the expenditure may be recovered and applied to further use may be considered novel. It happens only rarely that such a combination is feasible, one of the cases being that of the Homestake Mining Co., at Lead, S. D.

For the successful operation of the Homestake stamp mills, by means of which gold is recovered from the ores, a large quantity of water is required—more than is available in the near-by streams. To supply that water the company years ago built a pumping station at Hanna, on the far side of a divide, where a sufficient quantity of water is available. A steam plant was installed and Riedler pumping engines were used to force the water over the divide and into the Lead mills. The engines have been in use for 15 years, and not once has it been necessary to stop them in that time. Now, however, owing to the greater efficiency attained by utilizing steam through turbines and the improved facility of electrical transmission, it has been found advisable to discard the steam-driven pumps.

FIG. 1. EXTERIOR OF THE POWER STATION

The new units consist of three 2,000-gal., one 1,000-gal. and one 500-gal. double-suction turbine centrifugal pumps, directly connected to electric motors. The large pumps have 10-in. discharge pipes and are connected each to a 300-hp. 60-cycle three-phase motor operating on 2,200 volts and at 1,160 r.p.m., equipped with hand starters. The 1,000-gal. pump has 8-in. discharge and is connected to a 150-hp. motor of 1,750 r.p.m., and the 500-gal. pump to a 100-hp. motor having the same speed; the latter pump has a 5-in. discharge.

Because of the small amount of floor space required by the centrifugal pumps and their motors, it was found possible to assemble the installation on the lower floor, or basement, of the pumping station without disturbing the Riedler engines. Thus no interruption of the pumping service has been or will be required. At present one of the 2,000-gal. centrifugal pumps has been put into service, while foundations are being prepared for the others, and it will be only a short time until the service will be completely taken over by the new electrically driven pumps. The 2,000-gal. pump now running takes 250 kw. on the average, but there is no means for measuring the quantity of water delivered since there are two steam pumps still forcing water into the same discharge line. Power for operating the Hanna pumping plant is supplied electrically from a turbo-generating plant located at Lead, S. D.

From the pumping station the net lift for the pumps is 380 ft. Crossing the divide the water comes down grade on the Englewood side, having a total fall of 485 ft. It is here that the novel features of the arrangement become

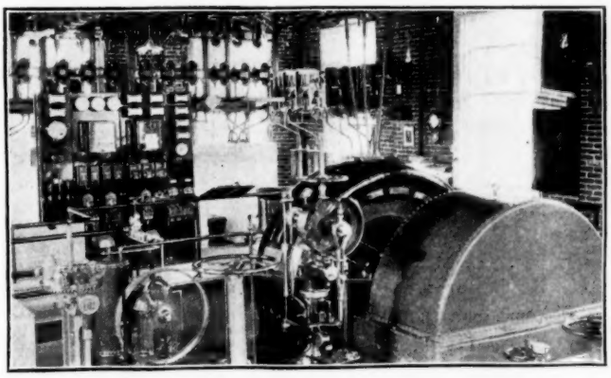


FIG. 2. ONE OF THE GENERATING UNITS

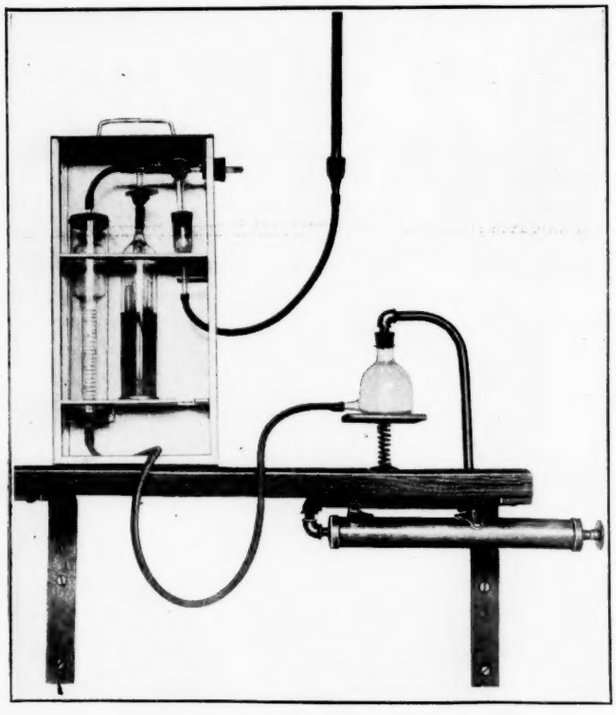
apparent, for the water column is led into the Englewood power-generating plant, where the energy of the water is converted into electrical current through Pelton wheels connected to electrical generators. Here a part of the power used in pumping the water over the divide is returned in a form available for further use. The generating station, Fig. 1, is equipped with the requisites for power generation and transmission. Fig. 2 shows some of the generating units. Through this plant about 400 kw. is recovered.

For the operation of the pumping station at Hanna, power is transmitted from the Lead power plant over a double transmission line, thus provided to prevent interruptions. The average work for the pumps is 700 minute-feet of water, and to this is added about 160 minute-feet of gravity water on the divide. The arrangement in its complete form makes a clever and unusual way of "getting one's money back."

Defender Gas Suction Pump

A device for cleaning the gas out of the pipe connection between boilers and CO_2 machines, called the Defender gas suction pump, is manufactured by the Defender Automatic Regulator Co., Oriel Building, St. Louis, Mo.

The pump illustrated herewith is mounted on a shelf and is connected to a water bottle on a spring-supported base that holds the bottle against a rubber stop. A couple



APPLICATION OF GAS SUCTION PUMP

of strokes of the pump plunger will clean the gas out of 50 to 100 ft. of ¼-in. gas pipe leading from the boiler. This permits of placing the CO_2 machine at a distance from the boiler, and the gas lines from the different units can be run to it through the rubber tube connecting with the vertical pipe. The appliance is suitable for use with any Orsat. It is a labor saver and there is practically nothing to get out of order or to wear.

JUST FOR FUN

Guards Find a Bomb

Plants engaged in making ammunition and arms in many instances employ guards to patrol their works. One of these fellows at a certain plant recently found what he took to be a bomb in a tunnel close to the works. It was a grim and deadly looking thing, and he had visions of promotion for finding it—that is, if it did not explode meanwhile. With the assistance of two other courageous guards, the bomb was carefully placed in a pail of water and soaked for several hours. Then it was removed and gingerly taken apart, but nothing of an explosive character was found, for, truth to tell, it was only a Sarco steam trap accidentally left in the tunnel.

That Powerful Gas

Barney's job for several days had been to break up the bundles of sheets for the "crown" of a 4,000,000-cu.ft. illuminating-gas holder and drag them to a point handy for the sheet-hanging gangs. He had gained a strong impression of the total weight of these sheets and near quitting time one day he looked up at the top of the guide frame some 150 ft. above him and said to a foreman: "Bill, will that crown go clear up to the top there, when they fill the tank, and if it does what takes it up there?" "The gas pressure," answered the foreman, "which will be about 6 oz. to the sq.in." "Only 6 oz.?" Barney was nonplussed. He knew the foreman could be trusted, but he thought of the sheets he'd dragged for a week. Then he looked across the giant tank some 200-odd ft. in diameter, back at the piled sheets still waiting, and then suddenly turned and delivered his solution: "Gosh! but that gas must be powerful stuff!"—H. B. McDermid, Worcester, Mass.

Holding the Water Level

Pat, the fireman in a certain power plant where the boilers were heavily loaded, was allowed to tend his own feed water for the first time. The rate of evaporation being pretty high, the feed pump had to be kept working constantly. After a time the engineer was conscious that the pump had stopped. He hurried to the boiler room, but a glance at the gage-glass showed that the water level seemed about right. "Pat, why isn't the pump running?" "Well, you see, sor, I've been havin' trouble all evening to hold enough wather, sor, so I just shut the little valve there, sor (pointing to the bottom of the glass), and now I can hold the wather level easy." The engineer jumped for the valve, opened it, found he could just barely avert a shutdown, and did so—but never again did Pat tend his own water.—H. B. McDermid, Worcester, Mass.

Cheap Pay and High Cost— . . . not mentioning the fact that generally cheap pay to attendants results in high cost per kilowatt-hour, usually on the coal item and often in maintenance.—Polakov.

Editorials

Latest High-Water Mark in Turbine Development

The development in magnitude and economy of the steam turbine has been such during recent years that somehow the profession is expectant—is immune to astonishment on the announcement of the latest "high-water mark."

At the time the thirty-thousand-kilowatt "cross-compound" units were developed for the Seventy-fourth Street Station of the Interborough Rapid Transit Company, their designer, Francis Hodgkinson, said in his article in *Power* of Sept. 15, 1914: "This cross-compound principle will permit of turbines being designed in what are, in effect, single units up to fifty or one-hundred thousand kilowatts, should machines of such large capacities be required. In the case of such sizes three stages would probably be employed." This prophecy is a reality, as an order for a seventy-thousand-kilowatt machine has been placed by the Interborough Rapid Transit Company with the Westinghouse Electric and Manufacturing Co. At this writing the water rates of the unit are not available, but they will be better than those for the Seventy-fourth Street thirty-thousand-kilowatt units now running, the best guaranteed performance of these latter being eleven and twenty-seven one hundredths pounds per kilowatt hour for a load of twenty-five thousand kilowatts. It is noteworthy that all three rotors of the new machine will run at fifteen-hundred revolutions per minute. A unit of the design of this largest one has advantages of control that are more than interesting. These will be treated of in an article now being prepared by one of *Power's* staff.

In these days, when so many are discussing the economical advantages of extremely high steam pressures, it is interesting to observe the results of the application of these theories to practice. In his capacity as consulting engineer for an electric company in Buffalo, N. Y., H. G. Stott sent out for tentative bids and guarantees for turbines to operate under two conditions: Two hundred and seventy-five pounds' pressure and two-hundred and seventy-five degrees' superheat; six-hundred pounds' pressure and superheat enough to approximately approach the same total temperature as in the first condition, namely, six-hundred and eighty-four degrees. In every case the guarantees for the more moderate pressure and higher superheat were appreciably better than for the higher pressure and more moderate superheat. Evidently then, when it comes down to "brass tacks," builders are not prepared, at this time at least, to guarantee what is so enthusiastically acclaimed by advocates of the very high pressure and moderate superheat unit.

If the chief reason is other than that the density of the steam at the higher pressure is considerably greater than at the lower, while the blade clearances remain constant, thus causing far greater leakage losses, it would be interesting to us all to have designers like W. L. R. Emmett and Oscar F. Junggren of Schenectady, Mr. Hodgkinson of Pittsburgh, Richard Rice, of Lynn, and others tell why.

The guaranteed water rate of the twenty-thousand-kilowatt machines for the Buffalo plant, where two-hundred and seventy-five pounds' pressure and two-hundred and seventy-five degrees' superheat will be used, is, as far as we are able to learn, the lowest on record for turbines, ten and two-tenths pounds per kilowatt hour.

Those who remember the astonishment, particularly among European engineers, when the two-thousand-kilowatt turbine was built for the Hartford Electric Light Company in 1900, because it was such enormous capacity to inclose in a single casing, will smile to know that each of the three thirty-five-thousand-kilowatt machines which the General Electric Company will furnish for the Fifty-ninth Street power house of the Interborough Rapid Transit Company will have a single casing. Distortion due to great temperature differences in a single casing seems to have ceased to be a bugbear, at least to the designer; but the man who operates a turbine knows that distortion is his nightmare.

Boiler Inspector's Main Concern

Engineers who regard the boiler inspector's visit to the plant with feelings of resentment are by no means rare. Sometimes, in such cases, the truculence displayed has its origin in excessive ego, which balks the operating man from getting an accurate mental focus on the why and wherefore of the inspector's visits. But generally it is because the engineer harbors a conviction in the depths of his soul, whether or not he is willing to admit the fact, that the boilers are in no condition to stand close inspection.

Whenever an engineer prates against the bringing in of an outsider to pass judgment upon the plant and gives full vent to his powers of persuasion in an effort to convince his employer that boiler insurance and the periodical inspections going with it stand for nothing more than an extravagant expenditure of money, it is a moral certainty that there is a robust Senegambian in the woodpile. There are some hidden facts about the boilers that do not comport with accepted principles of safety and economy.

It may be asked why these protesting engineers permit their boiler plants to run down to an extent that will make them dread the approach of the inspector. Why do they not put their boilers in such order that they will be proud to have the inspector take a look? The explanation may be seen only in the operating man's inertia, in his disinclination to get in motion when the only inciting impulse is his own consciousness that the job ought to be done. When some men are left to their own initiative in matters of this kind—when they are not goaded to their tasks under the critical eye of a superior—they are prone to lapse into habits of neglect, even in so serious a matter as the care of a steam boiler, and perhaps with a perfect understanding of the consequences.

The opponent of boiler inspection will perhaps aver that his education as a steam engineer, and particularly the experience that he has acquired during his term of service in the plant directly under consideration, renders him as well qualified as any man to know the state of the boilers and what activities are needed to keep them in a safe working condition. "I am as good a judge of a boiler's safety as any inspector on the job" is an expression frequently used by the engineer where boiler insurance is an issue.

Granting the soundness of this contention, that is not the point at issue. The need of the inspector's visit to the plant does not revolve around a question of whether the engineer in charge has a respectable knowledge of boilers and the ability to keep them in working trim. It might even be admitted that the engineer is fully the equal of the inspector in this regard; and for good measure, the inspector, if a complaisant sort of chap, might concede that the engineer outshines him a bit in his wealth of boiler lore. It is the fallibility, together with the flaw of unreliability latent in all human character in varying degree, that supplies one substantial reason—and for that matter, it might be put forward as the principal one—for projecting the boiler inspector into the field of power-plant management. He has no special concern either as to the depth or shallowness of the engineer's knowledge of boilers. His mission, first and last, is to search around and note if those things that make for the safety and preservation of the boiler structure are faithfully observed. The point of interrogation with the inspector is, in short, not what the engineer knows, but what he does.

⁂

Efficient Purchasing by the Engineer

Upon the engineer in many plants falls the responsibility of purchasing supplies, and unless he studies to buy wisely he is likely some day to become subject to criticism. The lowest price obtainable is not the solution of every purchasing problem; the best service for the outlay is the real criterion.

The engineer who would buy well today must know as much about the material as the man that sells it, and sometimes more. His knowledge must be broad, and it is seldom completed. His is a position demanding continual study, not only of new material or of that not before purchased, but of supplies repeatedly bought, so as to keep in close touch with conditions of sale and use. Competition should be encouraged even in minor purchases for one's employer. Only through competition can the engineer go into the market and buy with the certainty that the price paid is right. The compilation of a directory of supply dealers based upon information received from every source is helpful and above all, effort should be made to keep it uptodate.

Where a large list of possible bidders confronts the engineer dealing with a comparatively small order, the use of all the different concerns at a time may not be feasible, but enough should be selected to secure worthwhile bids, keeping open several possible sources of supply. The interest of the different competitors should not be allowed to lag, and the engineer will do well to avoid becoming dependent upon a small group of manufacturers. All bidders should be stimulated by expressions of thanks for their quotations, and also at times, on orders of any size, regrets should be extended for their unsuccessful proposals. This is a point often overlooked.

It ought not to be necessary to emphasize the importance of filling out complete orders with all needed data as to shipping directions, terms of delivery, price and signatures, keeping a copy for filing. An order is a contract between the buyer and the seller, and the seller is bound if he accepts it, even if he makes no formal acknowledgment to the buyer, to supply material of the stipulated kind and at the price stated. Great care should be exercised in issuing an order to have it call for material of the exact nature and quantity wanted. The custom of asking for supplies at the price last paid and without specifying the latter or putting the details of the material on paper opens the door to a great deal of possible delay and trouble.

There is no reason why the engineer should not also be a good business man within the limits of his field, and the exercise of common sense in careful purchasing will in the long run save the plant owner money and help prove the qualifications of the engineer to be intrusted with greater responsibilities.

⁂

If those responsible for Chicago's smoke were one-quarter as thorough in their work as was the technical staff that compiled the recent smoke report of the Association of Commerce Committee, the bound volume could be carried by at least one able-bodied man.

⁂

Firemen in Middle West plant use long-handled shovels to prevent burning their bodies when boilers are forced to 175 per cent. rating.—News item. Sometimes shirts are worn for the same purpose.

⁂

"Engineering is getting as popular as politics," says the president of Brown University. But Maurice L. Cook, of Philadelphia, can tell you that some engineers are not popular with politicians.

⁂

The coroner said he would not hold an inquest on the New Brunswick, N. J., boiler explosion in which two were killed. Well, it won't make any difference.

⁂

Every once in a while geologists tell us of the great quantity of coal in Alaska. What good will it do us? They will need it all to keep warm.

⁂

With a coal shortage staring us in the very bunkers, dealers in Bunkine to pour on the pile should do a great business.

⁂

We cannot help thinking that some day the small automatic refrigerating machine will put the ice man in storage.

⁂

And now the engineer plays King Winter to the public. Have you converted your freezing tank into a skating rink?

⁂

"Firing Tool Explodes."—Headline. Thus do the Fates pursue the man with the hoe.

Correspondence

Measuring Indicator Diagrams

I happened into a medium-sized steam-electric plant a few days ago and found the chief engineer working on one of his engines. An indicator was rigged up, and the chief was fussing and fuming over the cord attached to the reducing wheel. After a few trials to see how long a card was made, he would pull a pad from his hip pocket and laboriously do some calculating, shake his head and try again at the cord. After watching him a few moments, I inquired what was the trouble.

It developed that he had unpacked a new indicator a few days before and was attempting to take a few diagrams from this engine, which seemed to be working badly. He was trying to adjust the reducing motion and was at a loss to know what length the diagram should

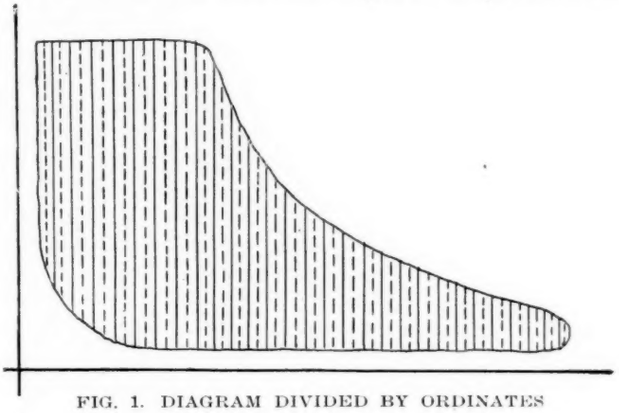


FIG. 1. DIAGRAM DIVIDED BY ORDINATES

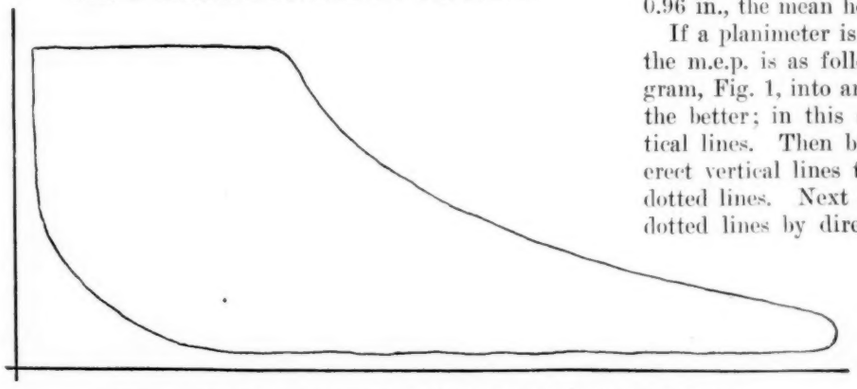


FIG. 2. DIAGRAM HAVING SAME MEAN HEIGHT AS FIG. 1

be. He said he knew that it had something to do with the revolutions per minute and the length of the stroke, but just couldn't get to it! He never had had to "break in" a new indicator, he said, as all the others he ever used "were set," and he never had this job to contend with before.

Of course I helped the old fellow out of his dilemma, but when I told him his card could be any length at all within the limits of the apparatus, he thought I was trying to fool him. There may be other engineers, first-class fellows like my friend, but who would also get caught with a new indicator.

After a diagram had been taken, the figure generally desired, after it shows that the valves have been properly set, is the mean effective pressure (m.e.p.) which is necessary in the horsepower formula.

The planimeter, when used, registers the area of the diagram. This area divided by the length of the diagram gives the mean height, or the average rise of the indicator piston during one stroke. The higher the pressure of the steam for a given scale of spring, the higher the piston will rise. After obtaining the mean height, multiply it by the scale of the indicator spring. By the scale of the spring is meant the amount of steam pressure per square inch that is required on the indicator piston to raise it 1 in.

Suppose that in Fig. 1 the diagram found by this method has an area of 2.88 sq.in., which is equivalent to a rectangle having a length of 3 in. and a height of 0.96 in., which latter is the mean height of the card. If, then, the scale of the spring used was 60, the m.e.p. during one stroke is $60 \times 0.96 = 57.6$ lb. per sq.in.

Going back to the point that was troubling my friend, the engineer, assume that the reducing motion was such that it described a diagram 4.5 in. long, as per Fig. 2; with the same scale of spring it is evident that the mean height of the new card is no more and no less than in the first case. Therefore by increasing the length of the card a certain percentage, we likewise increase the area in the same proportion. With the planimeter it will be found that the area of diagram No. 2 is 4.32 sq.in. Dividing this by the length of the card, or 4.5 in., we get 0.96 in., the mean height of the diagram, as before.

If a planimeter is not at hand, the method for finding the m.e.p. is as follows: Divide the length of the diagram, Fig. 1, into any number of equal spaces—the more the better; in this case 27, as shown by the light vertical lines. Then bisect each of these equal spaces and erect vertical lines through the points, as shown by the dotted lines. Next find the sum of the lengths of these dotted lines by direct measurement, but this is a slow and tedious process, as the length ought to be accurately scaled, which is not easily done.

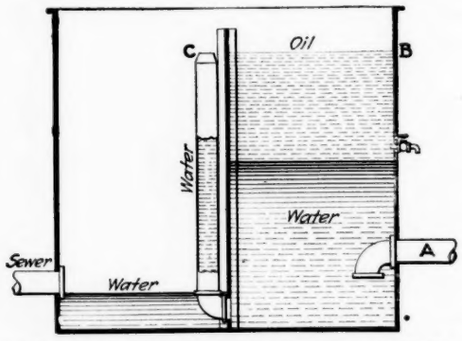
I recommend that a pair of dividers be used and each line stepped off on a continuous straight line; then measure the length over all of this line, which in the problem will be $25\frac{29}{32}$ in. This, divided by the number of dotted lines, will give the average length of all the lines, or 0.96 in., as before.

If the diagram were 4.5 in. long, the mean height would not change, which can be proved by taking Fig. 2 and dividing it by vertical lines as for Fig. 1. It will be found that the sum of the lengths of these lines is greater than the diagram Fig. 1, but dividing this sum by the number of spaces into which it has been divided, the average length of the lines will be 0.96 in., again showing that the length of the diagram is immaterial.

Erie, Penn. WILLIAM C. STROTT.

An Oil and a Money Saver

How many readers of *Power* are letting all small drips and drains from machinery run into the sewer? In almost all drips and drains there is a certain amount of oil that a separator like the one described and illustrated will catch, and I am sure that in the course of a year



OIL TRAP IN WASTE LINE TO SEWER

it will pay for itself many times over. In one plant the saving amounted to 100 gal. per month.

The separator is very simple in construction, being nothing more than a tank with a partition and a pipe connected in the bottom of the partition and extending up near the top.

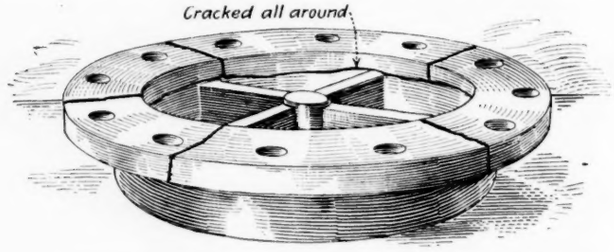
The action is as follows: A mixture of water and oil enters at *A* until the tank is filled up to *B*; then the water from the bottom runs over at *C* and thence to the sewer, and the oil is drawn off at the faucet. Once made, this oil separator is always on the job whenever necessary.

EDWARD C. EDNEY.

Birmingham, Ala.

Peculiar Cylinder-Head Break

The following account of a cylinder-head break may be of interest to readers of *Power*, as the location of the crack is somewhat out of the ordinary. The fracture occurred on the left cylinder of a pair of 250-hp. engines



HOW THE CYLINDER HEAD WAS CRACKED

direct-connected to a 400-kw. alternator and running noncondensing.

About 10 o'clock one evening the operator heard a slight click in the cylinder, which grew worse, whereupon he started another unit, but before getting the noisy engine shut down there was a loud knock and steam began to work out around the false head. As soon as the engine stopped, the false head was removed and the cylinder head was found to be cracked, as shown in the sketch and which is peculiar and was somewhat mysterious until the cause had been determined.

Upon removing the head it was discovered that the piston, which was solid and fitted with snap rings, had become loose. The piston was originally shrunk on the rod, countersunk about ⅜ in. and the end of the rod peened over. About three-fourths of the peened portion had broken off and dropped just far enough to become wedged between the cylinder head and the piston, the imprint of it being plainly shown on both. None of the studs was broken but all were bent outward. Luckily the head did not leave the cylinder.

Ithaca, N. Y. C. B. HUDSON

Should Ammonia Compressor Valve Be Closed?

An ammonia compressor was shut down each night, and the suction valve was closed and the discharge stop valves left open for safety when starting. The compressor discharge valve and piston leaked a little, and this put head pressure on the stuffing-boxes and resulted in considerable loss of ammonia, as the stuffing-boxes would not hold that pressure, though they would hold the back pressure when considerably slacked off.

Another compressor was shut down at night and the only main liquid valve from the receiver closed. There was some liquid in the coils on starting, but I believe that is better than putting head pressure on a stuffing-box not built for it. I prefer to close both suction and discharge stop valves if a shutdown is going to be long enough to make it worth while closing either. But it seems to me more important to get the discharge valve shut than to shut the suction if only one valve is closed. A rod will cool after the machine is stopped and will contract enough to leak with a high pressure on it even if the gland is tightened when first shut down. If the discharge valves are always shut and everyone knows it, they will be opened before starting. If not regularly shut, there is liability of someone starting with them shut.

Washington, D. C. MORRIS ELLISON.

Is There Power in Vacuum?

On two recent occasions at engineers' meetings which I have attended questions have been asked concerning vacuum, and there are those who seem to believe there is power in vacuum. My object here is to give my reasons for not agreeing with the advocates of this theory, and to prove that there is no power in vacuum.

Science teaches that each particle or space throughout the universe contains energy, and physics teaches that vacuum is space entirely devoid of matter. In creating a vacuum, mechanical means are employed to remove this ever-existing energy from one side of the equipment so that the atmospheric pressure can manifest itself on the other.

Some engineers consider magnetism and vacuum similar. This is a mistaken idea because the invisible and intangible lines of magnetic force will lift or pull iron or steel, but vacuum has no such power. Take a boiler or other closed vessel containing steam at any pressure above the atmosphere and allow the steam to condense, reducing its volume and creating a partial vacuum. On opening a connecting line there will be an inrush of air tending to equalize the external and the internal pressure. Energy is exerted through this inrush

of air and not through any efforts on the part of the vacuum.

A familiar experiment is that of a steam engine with the piston centrally located in a cylinder and a vacuum produced on both sides. "Produce" I believe is a more appropriate word than "create," for creating is building or making something from nothing, which is not possible, and in producing a vacuum we destroy or remove something. To the crank end is admitted atmospheric pressure, and the piston moves toward the head end with the inrush of air. This seems to be proof that vacuum has no power, the power being derived from the atmospheric pressure and not from vacuum.

It is true that with the use of a vacuum, an engine develops greater power. This is realized when operating an engine condensing and the vacuum breaks, thereby admitting atmospheric pressure against the piston. But there is usually some back pressure working against it and supposing this back pressure to be only 2 lb., it decreases the power of the engine. The 14.7-lb., or atmospheric, pressure will do precisely the same thing to a proportionally greater degree. Producing a vacuum is neither more nor less than removing this atmospheric pressure, which in this case is back pressure; and the more of this back pressure that is removed, giving the engine more freedom to work on the high-pressure side, the greater will be the power produced for the greater will be the mean effective pressure.

It therefore seems erroneous to say that there is power in vacuum, as we cannot produce effect without cause or movement without power. Inert matter cannot move itself, any more than can something be made out of nothing. The idea that such a thing as true power exists in a vacuum is misleading, as vacuum cannot pull any movable object, but it must depend entirely upon external properties for whatever power is derived from it, therefore there is no power in vacuum. L. E. KENNEY.

Brooklyn, N. Y.

Groaning in Steam Pump

E. Blondet, in the issue of Feb. 1, page 162, asks suggestions for the prevention of groaning in a steam pump. The cylinder oil used may be too heavy, and I would suggest trying a lighter oil. Since the pump has just been overhauled, it may be that the piston rings scrape the oil from the walls of the cylinder, in which case the remedy is to chamfer the edges of the rings slightly.

Another thing that may cause the groaning is tight packing on the water end of the pump. The excessive friction of too tight packing causes groaning, although that will stop after a while. If, however, he is sure the groaning is in the steam end, let him mix a little graphite with the cylinder oil and inject the mixture into the steam cylinder using a hand pump.

Middletown, N. Y. G. T. MICHAELS.

I once had a similar experience on taking charge of a new plant. The valves and steam cylinder showed good lubrication, and the piston rings were well fitted with good bearing surface all the way around and had a clearance gap of about 0.005 in.

On examining the water cylinder, I found the packing so tight that it was difficult to remove it with a packing hook. It had evidently been cut too long, for when it was cut a little shorter the groaning stopped.

I find it well to cut hydraulic packing short, about ⅛ in. to the foot, when used in a hot-water pump.

Saginaw, Mich. J. R. LACURE.

I also have read about Mr. Blondet's trouble. This is usually caused by one of two things—lack of lubrication or excessive rubbing action between parts. Two or three pints of good cylinder oil in 10 hr. certainly is enough, unless there is some way for the oil to escape without entering the pump.

From the fact that the pump cylinders were rebored and new rods supplied, I should imagine that the trouble was due to the cylinders not being in line as now assembled by the machinist. The pistons may be binding in the cylinders and the new rods acting the same way in the glands.

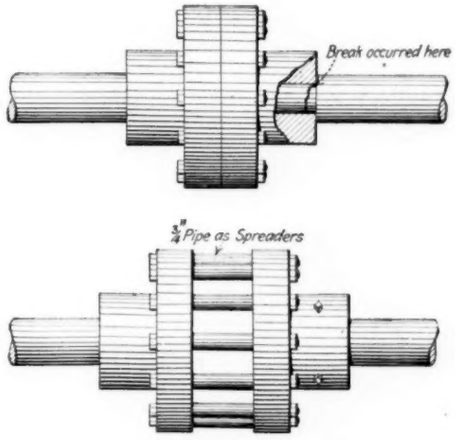
I would suggest that Mr. Blondet remove the pistons and rods from the pump and test the cylinders by stretching a thin line or wire through the glands from one cylinder end to the other and see if they are in line. He might also make sure that the piston rings are not too tight. In addition, a careful examination of the pistons, cylinders and rods will undoubtedly show where the rub is and, in fact, might reveal the cause of the trouble.

Brooklyn, N. Y. WARREN D. LEWIS.

Broken Fan Shaft Repaired

The illustration shows how a broken fan shaft on a lake steamer equipped with induced draft was fixed. The shaft broke flush with the hub of the coupling while the steamer was in midlake, and full speed could not be maintained with natural draft.

The broken piece of shaft was removed, the coupling was bored out in a lathe to fit the large part of the shaft,



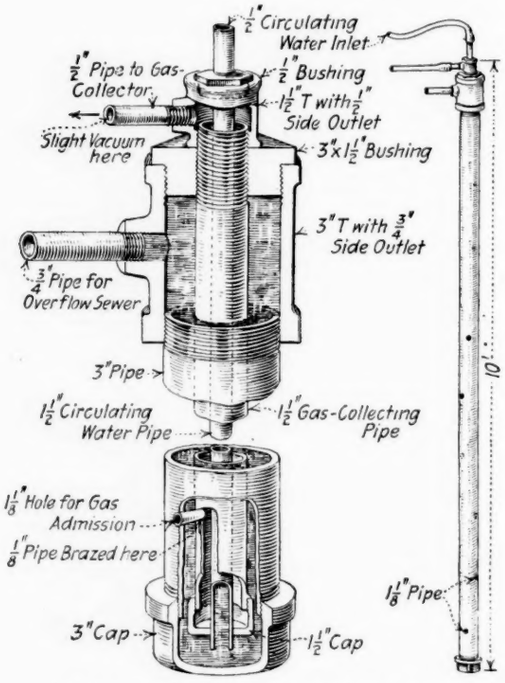
COUPLING ON SHORT SHAFT

and holes were drilled for two ¾-in. setscrews. Coupling bolts were made 3 in. longer than the original ones, and thimbles of ¾-in. pipe 3 in. long were cut to go over the bolts and act as spreaders. The work was completed in 2½ hr., and the boat was then able to proceed at full speed.

Toledo, Ohio. J. H. CUNNINGHAM.

Water-Cooled Sampling Tube

The sketches show a sampling tube that has been a success in taking samples of high-temperature gases. No ordinary pipe would stand the high temperature in some parts of the furnace. Provision is made for water circulation to the extreme end. The sampler is made of



WATER-COOLED GAS-SAMPLING PIPE

ordinary pipe fittings put together by ordinary methods, with the exception of the ⅛-in. gas-inlet tubes, which must be brazed—first, to the inner casing and, after inserting, to the outer casing.

For convenience, hose connections may be made to the inlet and outlet water pipes. A nipple-lined orifice is provided in the side wall of the furnace to admit the sampling tube. This nipple, extending as it does through the furnace wall, provides a suitable support for the tube. The nipple may extend sufficiently to be capped when not in use.

EDWARD T. BINNS.

Philadelphia, Penn.

Estimating on New Equipment

In estimating the cost of changing the location of machinery already installed or putting in new machinery, the accompanying tabulation will be found useful as a guide and a check on the work. Some form of this kind is essential in order to avoid omissions that may greatly influence the total of the estimate. The list submitted can of course be varied to suit the particular conditions. Under "Old Machinery" there may be added "Resale Price," in case the machinery is returned to its manufacturers for a cash consideration or is sold to some other concern. If the new machinery is made in the home shop, the burden, or overhead charges, should be included in its manufacturing cost. Certain items in the list may have to be omitted, as for example, if the driving power is taken from a lineshaft, there will be no estimate on numbers 6 and 7. Drawing charge is a very important item that is often overlooked in making up a preliminary estimate. Tentative drawings that have to be changed frequently take considerable time and are therefore expensive. In fact, the manufacturer of any type of machine should include in the selling price the drawing-room charges for the necessary layout and detail drawings, such as prospective customers may insist on having.

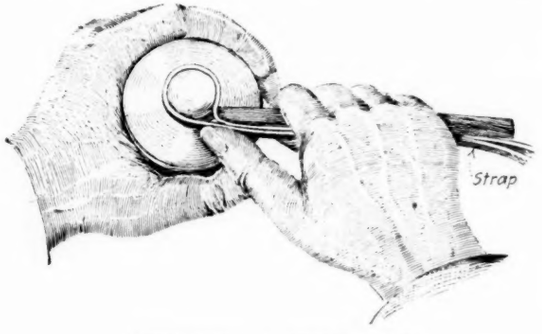
Walpole, Mass. D. FLIEGELMAN.

ESTIMATING COST OF NEW EQUIPMENT

1. Old Machinery—throwing out
2. New Machinery
   - a. Purchasing price, or
   - b. Manufacturing cost
     - 1. Stock
     - 2. Manufacturing
     - 3. Pattern work
     - 4. Drawings
3. Installing New Machinery
4. Foundations
   - a. Changing old foundations
   - b. Furnishing new foundations
5. Drive
   - a. Relocating old drive
   - b. New drive
     - 1. Shafting
     - 2. Hangers
     - 3. Clutches
     - 4. Pulleys
     - 5. Timbers
     - 6. Belting
     - 7. Motor supports
6. Motor
   - a. Relocating old motor
   - b. New motor (horsepower and speed)
7. Wiring
8. Piping
   - a. Changing old piping
   - b. New piping and valves
9. Drawings
   - a. Assemblies and details
   - b. Drives
   - c. Layouts
10. Buildings
    - a. Changing present buildings
    - b. New buildings
11. Excavating

To Loosen Small Metal Caps

A simple and excellent method for unscrewing small metal caps from cans is indicated in the illustration. Wrap a common shawl strap around the cap as shown, and "take a hitch" in it with your thumb or finger.



REMOVING A METAL CAP

Usually a very tight cap will let go immediately, without any trouble whatever.

If the cap is stubbornly tight, it can be "wrenched" off by using a wooden stick, as shown, on the side of the "hitch." A trial will prove its utility, especially with small sharply formed caps that might cut one's bare hands.

W. F. SCHAPHORST.

New York City.

A Hydraulic Jack should be filled with alcohol 1 part, water 2 parts, with a tablespoonful of sperm oil added. It should never be filled with water, kerosene or wood alcohol. Water is liable to freeze or rust the jack; kerosene destroys the packing; wood alcohol destroys packing and corrodes the metal.—"Engineering and Mining Journal."

The Lightest Oil used for lubrication has a specific gravity of about 0.865 and the heaviest about 0.930. The commercial range is from 0.885 to 0.907.

Inquiries of General Interest

"Calorific Power" of a Fuel—What is meant by calorific power of a fuel? K. S. V.

The term sometimes is applied, though inadvisedly, to the quantity of heat generated by the complete combustion of a fuel and to signify the same thing as the calorific or heat value, which is usually expressed as the number of B.t.u. that would theoretically be developed by complete combustion of one pound of the fuel.

Pump Water Cylinders Longer than Steam Cylinders—In a direct-acting steam pump, why are the water cylinders made longer than the steam cylinders? R. C.

To avoid excessive friction of the water in its passage to and from the water cylinders, the ports and passages of the water cylinders need to be much larger than those required for the steam cylinders which, for diminishing the clearance space to dimensions consistent with construction and operation of the pump, are reduced to a minimum to obtain greater steam economy. In either case the cylinders must be long enough to include the ports that are required beyond the strokes of the pistons, and therefore the water cylinders must be longer than the steam cylinders.

Differences of Steam Tables—Why are the B.t.u. values that are quoted in the older tables of properties of saturated steam different from those given in more recently published text and reference books? H. T. C.

The B.t.u. values given in the earlier tables generally were based upon interpretation and modifications of the results of Regnault's experiments, using as a British thermal unit the quantity of heat required to raise the temperature of 1 lb. of water 1 deg. F. at or near its temperature of maximum density, which is usually assumed to be 39.1 deg. F. The steam tables more generally used at present by engineers of the United States are those of Marks and Davis, based on more recent researches, such as those of Henning, Knoblauch and other authorities, reduced to a "mean B.t.u."; that is, $\frac{1}{180}$ of the heat needed to raise one pound of water from the freezing point to the boiling point. This value was used because of the variation of the specific heat of water between temperatures of 32 deg. F. and 212 deg. F. The differences between the older and newer tables, in quotations of the total heat of saturated steam at usual pressures, are generally within a fraction of 1 per cent.

Velocity of Water in Suction and Discharge Pipes—What would be the velocity of the water in the 6-in. suction pipe and 4½-in. discharge pipe of a pump having 10x12-in. water end, when running at 60 double strokes per minute with 10 per cent. slippage? E. R. G.

With a stroke of 12 in. the piston speed of the pump in making 60 double strokes per minute would be $\frac{12 \times 60 \times 2}{12} =$ 120 ft. per min. Without pump slippage, if either of the pipes was of the same diameter as the water cylinder, the velocity in that pipe would be the same as the piston speed of the pump, and for any other size of suction or discharge pipe, the velocity would be inversely as its cross-sectional area to the cross-sectional area of the pump cylinder. As the cross-sectional areas would be as the squares of their diameters, then with 10 per cent. slippage the velocity in a 6-in. suction pipe would be

$$120\ (100 - 10) \times \frac{12^2}{6^2} = 432 \text{ ft. per min.}$$

and in a 4½-in. discharge pipe the velocity would be

$$120\ (100 - 10) \times \frac{12^2}{4.5^2} = 768 \text{ ft. per min.}$$

M.e.p. and I.Hp. for Given Cutoff Depend on Diagram Factor—What would be the indicated horsepower of a 20x24-in. engine running noncondensing at 150 r.p.m. when supplied with steam at 90 lb. gage and cutoff at ⅜ of the stroke? W. L.

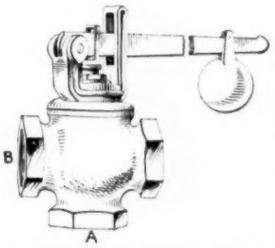
The mean effective pressure and power developed would depend on the diagram factor or ratio of area of the actual indicator diagram to the theoretical diagram that would be obtained with the same initial pressure and cutoff, without any rounding off at the points of admission, cutoff or release and with back pressure at atmospheric pressure. The ratio of the actual to the theoretical diagram would vary according to the type of engine, percentage of clearance and the distribution of the steam, as determined by the size of steam passages, leakage of piston and valves and the valve setting. Assuming that the mean effective pressure realized is 35 lb. and omitting consideration of the reduction of the piston area due to the piston rod, the engine would develop

$$\frac{35 \times (20 \times 20 \times 0.7854) \times \frac{24}{12} \times 150 \times 2}{33{,}000} = 199.9,$$

or practically 200 i.hp.

Openings of a Cross Safety Valve—What is the purpose of providing a safety valve with so many openings as shown in the sketch? J. W. C.

A safety valve having this type of body, from its resemblance to a cross fitting is known as a cross safety valve. The partition, or diaphragm, on which the valve seats is made of such form that the bottom opening A and one of the side openings, as B, connect with the pressure side of the valve. The other side opening is connected with the space above the diaphragm and is the relief outlet. At one time, safety valves of this kind were extensively used on boilers and other pressure vessels, having a single outlet that was usually connected to the bottom outlet A, while the communicating opening B was utilized as a service connection, but this use of a safety valve is likely to interfere with its proper operation and is inconsistent with good practice. The principal advantage of the cross safety valve is in affording the choice of a horizontal or vertical inlet, but under any circumstance worthy of obtaining certainty of operation of the safety valve, the inlet not required for operation of the valve should be plugged.



CROSS SAFETY VALVE

Pressure of Water per Foot of Head—Knowing the height, or head, of water above a given point, how is the pressure per square inch determined? P. K.

For nearly all practical computations the compressibility of water can be neglected, and when the water is of uniform temperature its density may be considered to be uniform. The pressure at each horizontal layer, due to the weight of the water, may therefore be considered as directly in proportion to the depth below the surface. Hence, in determining the pressure per square inch due to a given height, or head, of water, it becomes necessary to decide on the density or weight per cubic foot. This varies according to the temperature. Referring to the table of "Density and Volume of Water" given on pages 542-3, Oct. 19, 1915, issue of "Power," it is to be seen that the density varies from 62.4201 lb. per cu.ft. at 32 deg. F. to 59.6820 lb. at 218 deg. F.

As shown by the table referred to, the density from 50 deg. F. to 70 deg. F. varies from 62.4144 lb. to 62.3034 lb., or only about ⅛ of 1 per cent., but for convenience in calculations, for ordinary temperatures the density is usually assumed to be 62½ lb. per cu.ft. Assuming this weight per cubic foot, the pressure per square foot for each foot of depth would be 62½ lb. and the pressure per square inch would be 62½ ÷ 144 = 0.4328484 lb., or, as commonly taken, **0.433 lb. per sq.in.** for each foot of depth. Hence for ordinary temperatures of standing water, the pressure per square inch due to the height, or head, of the water above the point considered is usually obtained by multiplying the head in feet by the constant 0.433.

[Correspondents sending us inquiries should sign their communications with full names and post office addresses. This is necessary to guarantee the good faith of the communications and for the inquiries to receive attention.—Editor.]

The Prevention of Corrosion in Pipe Lines*

By F. N. Speller†

SYNOPSIS—Experiment and observation by the author tend to show that corrosion of pipe is more a matter of the amount of oxygen in the water than of the composition of the metal. After rust is formed, corrosion is accelerated. There is three and a half times as much oxygen in solution in water at 120 deg. F. as at 180 deg. F. Means are suggested for deaërating water.

Casual observation will show marked differences in the degree of corrosion of pipe in service. For instance, hot-water heating systems and sprinkler systems show practically no deterioration in service after 25 years, while low pressure steam returns sometimes give trouble after 15 years' service or less. Galvanized pipe in hot-water supply systems, where the water is heated under pressure, lasts from about 5 years upward, depending on the temperature and quality of the water and the volume of flow. The last-named condition is so severe on iron and steel pipe that many are compelled to use brass pipe, at a cost approximately ten times that of galvanized. These few instances are the extremes, but are surely suggestive when we consider that in pipe carrying ordinary water under some conditions there is no apparent deterioration in a generation; whereas in other cases, the same grade of pipe is seriously damaged in a very few years.

Some years ago, when steel pipe was comparatively unknown and not fully developed, it was natural to question this material, but comparisons of modern wrought-iron and steel pipe in the same lines in service have shown beyond any question that where corrosion is found one material suffers on the average as much as the other.

It has been the custom of the writer to keep several service tests under way continuously for the past several years. In every case the coöperation of some local engineer or organization is sought under whose immediate direction the test is conducted. One of the most recent to be completed may be described as an example of the method pursued in conducting such tests. This test was made in the Pennsylvania Building, Philadelphia, Mr. Munsey, engineer. Four standard grades of pipe of well-known manufacture were selected, and four pieces of each taken at random and coupled together alternately so that the hot water passed through each sample at the same temperature.

Date installed—Oct. 27, 1913.

Date removed—Nov. 20, 1915.

Location—Hot-water return line Pennsylvania Building, Philadelphia.

Temperature of water—175 deg. F. average.

Amount of water—5,000 gal. per day for 600 days.

Method of installation—In form of box coil.

TABLE 1. CORROSION MEASUREMENTS

Material	Sample Number	Average of 10 Deepest Pits in Each Piece, In.	Deepest Pit, In.
Steel	1	0.068	0.094
Steel	2	0.087	0.124
Steel	3	0.045	0.050
Steel	4	0.063	0.074
Steel	5	0.079	0.104
Steel	6	0.056	0.094
Steel	7	0.054	0.065
Steel	8	0.070	0.081
Iron	9	0.065	0.073
Iron	10	0.072	0.097
Iron	11	0.078	0.105
Iron	12	0.077	0.080
Iron	13	0.055	0.075
Iron	14	0.067	0.112
Iron	15	0.067	0.078
Iron	16	0.073	0.090
Steel	General Average	0.066	0.085
Steel	General Average	0.067	0.086
Iron	General Average	0.073	0.089
Iron	General Average	0.066	0.088

A résumé of a number of these service tests compiled from several sources is given in Table 2.

Inasmuch as both materials have been found to fail in about the same time under the same conditions, and as both have shown practically no deterioration after many years under other conditions, it would seem that with a correct understanding of the fundamental causes of corrosion a practical solution of this problem should be possible, especially after the considerable study given corrosion.

The inside of pipe is subject to peculiar conditions not to be compared with external corrosion, and the inside surface is particularly vulnerable in that protective coatings are difficult to apply, therefore are more liable to be defective and must, therefore, be closely watched.

Consider the situation in a hot-water heating system and a hot-water supply system where the temperature of the water is about the same. Evidently the water alone is not responsible for the results observed, but rather something brought in with the water. The hot-water heating lines have started to rust, and then the action has apparently stopped, while in hot-water supply lines the action is continuous and rapid: so much so that if the pipe does not fail by leaking, it may be plugged tight with the reddish hydroxide of iron. The only way to account for this accumulation of oxide of iron is through the oxygen in solution in the cold feed water, amounting to 6 to 10 cc. per liter, according to the temperature and quality of the water. This small percentage of oxygen is apparently the measure of the destructive power of the water and accounts for the fact that a limited volume of water has no serious action on iron, whereas when this water is renewed continually, especially when heated, the results are liable to be disastrous. It will be useful to consider the mechanism of corrosion before discussing ways and means for preventing it.

All water supplies carry more or less foreign matter in solution. What are usually considered the purest natural waters are generally saturated with oxygen and carbonic acid, which cause such waters to be corrosive, particularly when heated. Iron in all its forms is soluble in water to the amount of a few parts per million, depending on its composition and that of the water. In this paper, in referring to water in connection with corrosion, it will be understood to include domestic supplies of the usual degree of purity. Chemically pure water does not occur in nature, and therefore may be omitted.

The phenomenon of solution is now generally explained as an electrochemical reaction. When pieces of zinc and copper are connected together and suspended in water, a current of electricity starts to flow through the water from the zinc to the copper. The zinc is termed the anode and

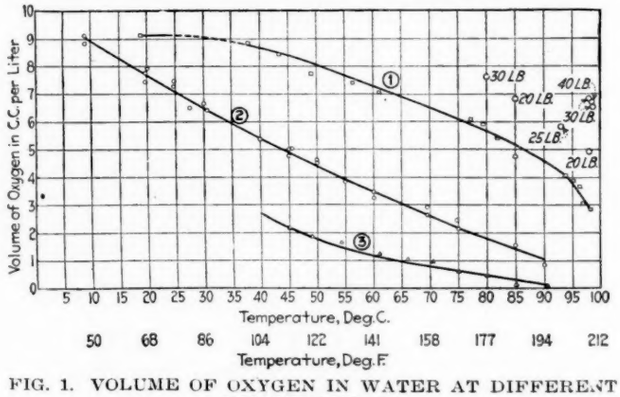


FIG. 1. VOLUME OF OXYGEN IN WATER AT DIFFERENT TEMPERATURES

the copper the cathode. While the current flows the zinc goes into solution, the amount dissolved being proportional to the current, according to Faraday's Law.

If we replace the zinc with a piece of iron, a current flows in the same direction and iron will be found in solution. Suppose we now replace the copper with another piece of iron. A small current of electricity will still flow, but not necessarily in the same direction, this depending upon the relative surface condition of the two pieces of iron.

It is this small current flowing between one piece of iron and another under water which causes iron to go into solution, and this is now recognized as the initial reaction of corrosion. Solution is hastened by carbonic acid and mineral

*Abstract of paper before the American Society of Heating and Ventilating Engineers, annual meeting, January, 1916.

†Metallurgical engineer, National Tube Co.

salts in solution, as these make the water a better electrolyte. However, it has been proved that iron will dissolve to some extent in the purest water that has yet been made. If the iron is exposed to nothing but water, this reaction will soon cease, owing to the accumulation of hydrogen at the cathode causing polarization; and this is what actually happens in hot-water heating and other systems in which the water, and consequently the supply of free oxygen, is not renewed. On the other hand, when oxygen is present it combines with the hydrogen, depolarizing the surface of the iron, causing solution of the iron to continue. Oxygen enters further into the

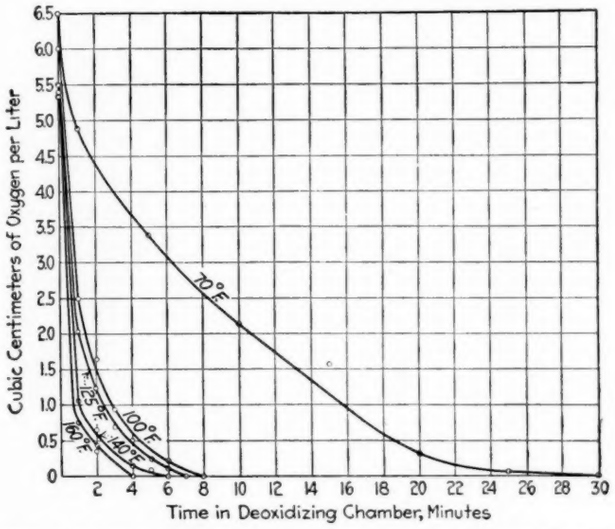


FIG. 2. DECREASE IN OXYGEN IN WATER AT DIFFERENT TEMPERATURES WHEN SUBJECTED TO DEOXIDIZING TREATMENT

reaction by combining with the ferrous hydroxide to form insoluble ferric hydroxide, generally known as rust. With an unlimited supply of water and oxygen, corrosion will continue until the iron is all converted into the form of ferric hydroxide.

So far most of the authorities are agreed as to the cause of corrosion, although there has been considerable scientific argument as to whether a trace of carbonic acid (CO_2) is necessary for the solution of iron. For all practical purposes we can let this question rest and combine the acid and electrolytic theories into one, as outlined, which affords the best explanation of the observed facts available at this time.

Ever since the electrochemical theory of corrosion was proposed by Whitney in 1903, there has been a division of opinion as to the cause of the difference of potential observed between two pieces of iron. The majority at first assumed that this was due to variation in composition of the metal, and the manufacture of iron of a high degree of purity in the openhearth furnace was heralded with great expectations as to durability. So far, after several years of trial it has not been found that such iron is so well adapted for the manufacture of pipe as the grade of soft weldable steel now generally used for this purpose.

In 1904 the writer started a study of the potential differences as found on the surface of iron of various compositions and has invariably found just as much difference in potential on the surface of very pure iron as on steel or wrought iron of ordinary commercial quality. Subsequent observations, covering several years of service with pure iron, openhearth and bessemer steel and wrought iron of the quality required for the manufacture of wrought pipe, have confirmed the conclusion expressed by the writer after his earlier experiments; namely, that composition has very little to do with the rate of corrosion of these metals under water. It should be remembered, however, that conditions underground or inside pipe lines are not the same as when exposed to the atmosphere, so that these conclusions do not apply to materials subjected directly to atmospheric conditions, such as metal roofing.

The tests and experiments referred to indicate that differences in finish and density of the material, particularly the character of the mill scale and how firmly it is attached, usually determine where corrosion starts and how it proceeds. The difference of potential due to surface influences was found to be many times greater than that due to variations in composition in the ordinary run of steel and predominated over all other influences in nearly every case. These conclusions were checked in the most critical manner and have since been borne out by service tests of several years' duration.

Among the surface influences that directed the course of corrosion, it was found that rust, when once formed, was nearly as potent as mill scale in its effect on corrosion. Some recent work by James Aston, of the United States Bureau of Mines, confirms these conclusions, but goes further in showing that the influence of rust in some cases is to render the metal underneath the rust anodic. As the mill scale is always the cathode, there is every reason to believe that we may have in certain places on the surface nearly double the difference of potential that was expected and this without reference to the actual composition of the iron or variations thereof. Everything seems to point to this explanation of the cause of pitting as being the true one. Under some conditions in water lines or boilers, it frequently happens that the tubes that are of high quality as regards chemical composition, structure and physical properties have rapidly pitted through in places. The difference of potential and the current which thereby flowed from the exposed places to the firmly attached mill scale, especially after the exposed metal became covered with rust, affords a satisfactory explanation of the rapid pitting observed in such cases.

TABLE 3. TYPICAL READINGS BETWEEN PLATES OF SAME AREA

Submerged Area of Plate in Sq.In.	Current in Milliamperes	Current Divided by the Square Root of the Area of Plate	Distance Between Plates, In.	Material of Plates
0.155	0.07	0.18	1	Clean soft steel plate free from scale (positive).
0.338	0.10	0.16	1	Same with light coating of mill scale (negative).
1.643	0.25	0.20	1	
3.875	0.48	0.24	1	
16.430	0.98	0.24	1	
0.062	0.06-0.09	...	⅛	Plug of pipe steel (positive). Similar area on skelp with light mill scale (negative).
0.342	0.14-0.18	...	⅛	
16.430	2.02	...	⅛	

The current flowing between two points on the surface of a piece of iron is very difficult to measure accurately; however, we have frequently observed currents as high as one milliampere flowing between steel electrodes in McKeesport city water, one plate being clean and the other covered with mill scale. Table 3 gives some typical readings between plates of the same area made with a milliammeter having an internal resistance of eight ohms; electrolyte, McKeesport city water.

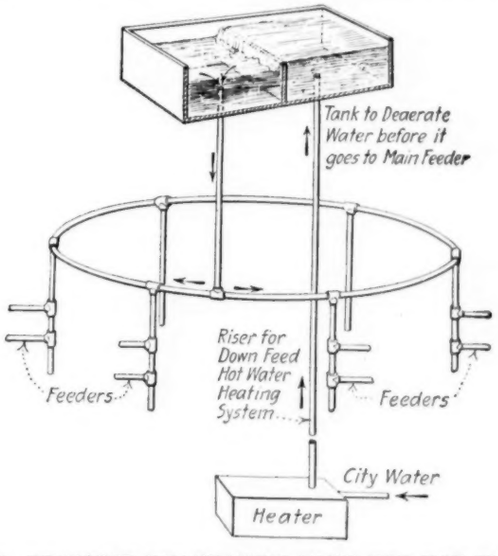


FIG. 3. PROPOSED DEAERATING TANK FOR HOT-WATER HEATING SYSTEMS

It might seem at first that these currents are too small to cause serious damage. A rough calculation based on these experiments will indicate the ultimate result from such currents acting continuously with certain submerged areas of electrodes; 1 ampere acting for 6 mo. will dissolve 10 lb. of iron, or a plate 12x12x¼ in. thick; 1 milliampere acting for 6 mo. will dissolve 0.144 sq.in. of this plate, making a hole about $\frac{1}{16}$ in. in diameter by ¼ in. deep; 0.1 milliampere acting for 60 mo. will perforate this plate with a hole of the same

size. This rate of pitting is not so far different from that experienced under some conditions of service.

The remedy seems to lie in the elimination of dissolved oxygen from water before use. This may be accomplished in at least two ways:

The first is by allowing the hot water to come to rest for a few minutes under greatly reduced pressure. As no reliable data could be found on the amount of oxygen retained in solution in water at various temperatures and pressures, a series of experiments were run to determine these constants. The results are given in Fig. 1. Evidently the pressure must be reduced below normal or the temperature raised nearly to the boiling point with the water at atmospheric pressure, to get proper separation of oxygen and other gases.

Some 2-in. wrought iron and steel pipe (galvanized) after thirteen years in the Columbia Baths, Atlantic City, N. J.—an "open" heating system—is in good condition. Copper pipe used under the same conditions in a "closed" heating system lasted only six months. The salt water was heated to 180 deg. F. in an open tank from which it was run by gravity to the system. This pipe was in continuous use during the season for this period. The galvanized pipe was practically as good as new, showing that even salt water is practically harmless when deaërated to this extent.

An alternative method of reducing corrosion in water lines by satisfying or "fixing" the free oxygen was tried out by the writer several years ago, using clean steel turnings. It was found difficult to get the scrap free from oil, and after rusting had progressed for some time there was a tendency for the mass to cake together and so impede the flow. By using sheet iron so formed as to provide a large number of channels with ¼-in. clear passage through which the hot water slowly percolates, we expect that these difficulties will be overcome. Some determinations of the rate at which the free oxygen is removed from water by this system are given in Fig. 2. The rate of rusting varies with the surface condition of the plates, becoming more rapid as the surface is covered with a good coating of oxide. The speed of rusting is 50 per cent. more rapid at 110 lb. pressure than at atmospheric pressure, and of course the time required to "fix" the free oxygen of the water varies with the amount of surface of metal exposed per cubic foot and other conditions that are liable to vary. For these reasons the results given in Fig. 2 are only relatively correct for the conditions stated.

On this principle two small plants have been equipped to carry out this method of treatment in practice. These systems were installed at places where considerable trouble has already been developed through the clogging and corrosion of galvanized pipes.

These plants, which have only been operating a short time, show a reduction of oxygen contents from 8 or 9 c.c. per liter to 0.1 to 2 c.c. per liter according to the rate of flow and temperature (Fig. 2). At present it seems desirable to design the plant so that the oxygen contents will be less than 1 c.c. per liter at all times, at which point corrosion seems to be reduced to a negligible amount. Some more definite data on this point will be available after these plants have been in operation for several months. The indications are that the rate of rusting of the plates, and hence the efficiency of the apparatus, will increase with time. Water should be in contact with the plates for at least six minutes.

Similarly, the corrosion of low-pressure steam lines will be found to depend principally on the amount of oxygen that finds access to the system. The return lines naturally suffer the most and are usually the first to show failure. Condensed water, when freed from oxygen in solution, is harmless and will not even tarnish bright iron after months of exposure; but this water, on account of its great purity, has greater capacity for solution of oxygen than the average natural water and is therefore apt to be very corrosive when aërated. This may be prevented in large measure by using an open feed-water heater and keeping the water over 185 deg. F.

[Mr. Speller added that he thought it would be well-worth while to provide a tank at the highest point in a down-feed hot-water system to deaërate the water before letting it into the feeder main or loop. The layout would then be as shown in Fig. 3.]

In some cases the surface of the pipe may be protected with a film of oil deposited from the steam. The writer's attention was recently called to the satisfactory results obtained in some buildings using exhaust steam, which on investigation were accounted for by the thin film of oil found on the inside of these pipes. This was such an interesting matter that we made some tests in the research laboratory on long lines of new pipe and found that mineral lubricating oil, when dropped at the rate of two drops per minute into a pipe carrying steam under pressure was carried forward in a fine state of division and in a few minutes was found condensed in a uniform film in the pipe, 160 ft. from the lubricator. While this simple means of protection would perhaps be objectionable in some cases, there are many steam-heating systems where the oil could do no harm and might result in considerably prolonging the life of the lines. Of course nothing but a good grade of mineral lubricating oil should be used, and the supply should be regulated by a reliable sight-feed lubricator.

Pipe for steam heating under certain conditions is subject to rather rapid deterioration from inside corrosion due to the large amount of water condensed in the returns. A case of this kind was investigated by the writer last summer, where corrosion had developed on the low-temperature end of the system owing to oxygen brought in with the feed water. The same plant had been operating for fifteen years without trouble, using an open hotwell kept at high temperature. Owing to enlarged demands on the system, certain changes were made by which the temperature of the water from the hotwell, from which the feed pumps drew their supply was reduced below 120 deg. F. At this temperature the amount of oxygen in solution is about 3½ times the amount found in water at 180 deg. F., most of which was of course absorbed in the water of condensation in the returns, making possible the corrosion of the pipe. A higher temperature in the hotwell and the use of a lubricator were recommended.

High-Temperature Insulation

On Dec. 14 the Chicago Section of the American Society of Mechanical Engineers held its second dinner of the year at the Hotel La Salle. The subject was High-Temperature Insulation, on which P. A. Boeck, chemical engineer and district manager of the Kieselguhr Co. of America, New York City, gave an interesting talk illustrated by numerous lantern slides. The speaker referred to the work that has been done in this field. He enumerated the advantages of insulation and briefly explained the mechanics applying to heat flow. He pointed out that heat is a form of energy consisting of molecular vibration of a periodic character and subject to the general laws of wave motion. It can therefore be reflected, refracted and dispersed. In other words, by proper mechanical manipulation the rate of flow can be increased or decreased within certain limits. One of the most common ways of doing this is by introducing alternate layers of materials having different heat-transmitting values, thus breaking up or changing the wave length. The introduction of air cells or voids within the body is a most effective way of reducing conductivity. The density of a body may be taken as an approximate criterion of its heat-conducting capacity. Large voids, however, have the effect of propagating heat by convection and radiation.

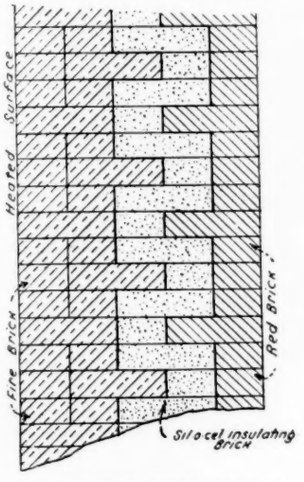


FIG. 1. INSULATION BRICK LAID IN WALL

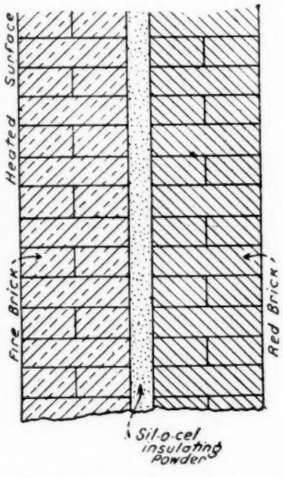


FIG. 2. INSULATING POWDER IN HOLLOW WALL

Heat is transferred from one place to another by three different methods: conduction, radiation and convection. Conduction is the action taking place in the transfer of heat in which the heat energy is passed along from a particle at a higher temperature to a lower one by virtue of their contact. This method of heat transfer is only of magnitude in solids. By radiation heat is transferred from one body to another

without any material agency by wave motion. It applies only to the gaseous state of matter. Convection is that process of heat transfer in which some portion of the body whose temperature is raised moves to another place in the medium where the temperature is lower, thus tending to raise the temperature of the medium. This method of heat transfer is therefore limited to liquids and gases.

In a steam boiler the heat generated by the fuel burning in the firebox is imparted to the dry surface of the boiler plate by radiation from the hot fuel bed, furnace walls and luminous flames, and secondly from the moving gaseous products of combustion by convection. From the dry surface it travels through the soot, metal and scale to the wet surface purely by conduction and is in turn transmitted into the boiler water mainly by convection. That part of the heat transferred to the walls of the furnace by any of the three methods mentioned is conducted through the setting and lost by radiation or convection from the outer surface of the setting if preventive means are not provided. It is the heat lost in this manner and the methods for its prevention to which the speaker gave special attention.

The amount of heat conducted through a unit area from one part of the body to another is proportional to the difference in temperature of the two parts—directly proportional to the thermal conductivity of the body through which the heat passes and inversely proportional to the distance between the two parts of the body. Heat transferred from one body to another by radiation is proportional to the difference of the fourth powers of the absolute temperatures of the two bodies. While this is strictly true only of the ideal "black bodies," the variation is so small that for all practical purposes the relation holds good in ordinary procedure. This relation indicates why hollow wall space is an effective insulator in low-temperature work such as refrigeration, whereas in high-temperature operations the loss of heat by radiation through a hollow wall space is so great that its insulating effect is less than if this wall space were filled with material of rather high thermal conductivity. This is especially true if the air space in the hollow wall is near the furnace side and becomes highly heated. This is contradictory to the general belief that since air is a poor conductor of heat, air spaces built into the walls of a furnace will greatly reduce heat loss by radiation. While the heat does travel slowly through the air by conduction, it leaps over the air space readily by radiation, because the quantity of heat that passes across the hollow space is a function of the fourth power of the absolute temperatures of the surfaces inclosing it, which loss is enormously increased by rise in temperature.

The speaker outlined the requirements of insulators, gave an indication of the amount of insulation required and illustrated the use of a formula developed by C. H. Herter to calculate the heat flow and temperature gradient through a furnace wall. A detailed description of the mineral known as "Celite," which is the product used by Mr. Boeck's company in block or powdered form for insulating purposes followed. This mineral is of a highly siliceous composition and of light weight. It is found on the Pacific Coast in an exceptionally pure state. In its natural rock form, air dried, it weighs from 25 to 30 lb. per cu.ft. When this material is ground properly so as not to destroy the cell structure, it weighs but 8 lb. per cu. ft. and has a thermal insulating power about equal to that of cork, or nine to ten times the insulating power of ordinary firebrick. Being almost pure silica, its melting point is high (2,930 deg. F., as reported by the Bureau of Standards) and it can be subjected to high temperatures without fear of alteration.

The product is prepared in brick or block form of various sizes and shapes by sawing the natural material. In crushing strength these bricks withstand over 400 lb. per sq.in. and are sufficiently strong to stand transportation and handling. Their cost is but little more than that of firebrick, and the cost of the powder is about one-third as much.

Numerous illustrations of "Celite" mines and the method of removing the product were shown. Many slides followed showing typical wall insulation, of which there are four forms of construction for high-temperature insulation which can be adapted to almost any character of equipment. Fig. 1 indicates the usual manner of using "Sil-O-Cel" bricks interlaid between a course of firebrick and red brick to prevent heat leakage through the walls. This form of construction is largely used in boiler settings, reverberatory furnace walls, roofs, etc. Fig. 2 indicates an insulating wall in which an otherwise hollow space is filled with insulating powder; 2 to 4 in. is usually sufficient. The powder is packed lightly to a density of from 12 to 15 lb. to the cubic foot, at which point it is not subject to settling or contraction due to either vibration or heat.

Figs. 3 and 4 indicate the method of insulating brick walls that are already in place. This form of insulation can be applied to old construction as well as new. In Fig. 3 expanded metal lath is erected on angle iron at the required distance from the outer wall and covered on the outside with one or more coats of portland-cement plaster, to which a small amount of "Sil-O-Cel" powder, approximately 20 per cent. by volume, has been added to give greater plasticity and ease of working and to increase the heat-resisting properties of the cement. The "Sil-O-Cel" powder is packed between the brick

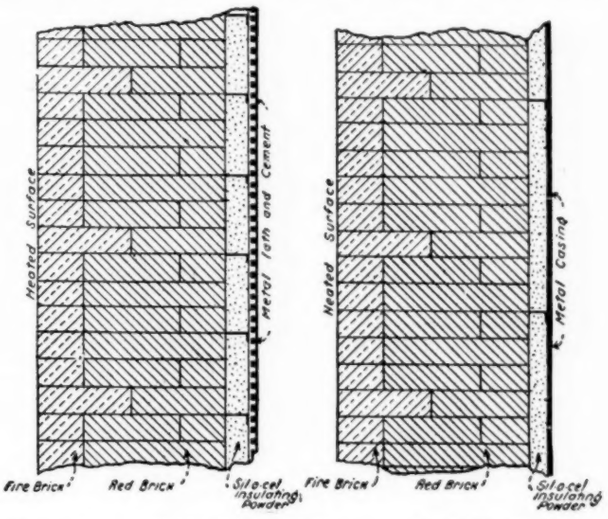


FIG. 3. POWDER SUPPORTED BY METAL LATH

FIG. 4. POWDER SUPPORTED BY METAL CASING

wall and the expanded metal lath. In Fig. 4 the powder is packed between a metal casing and the brick wall. The insulation of hot-blast stoves, metallurgical and electrical furnaces was given brief consideration.

In the discussion following the lecture, Joseph Harrington spoke highly of "Sil-O-Cel" products as insulating material. He had already applied the brick to a couple of installations with highly economical results.

Professor Richards, of the University of Illinois, was particularly interested in the formula derived to calculate heat flow through a wall. The value of the formula depended upon the accuracy with which its coefficient had been determined. At present the coefficients are far from accurate, and a great deal of experimental work is necessary to obtain even approximate values.

Colonel Henry Allen stated that the greatest trouble with insulating material was the commercial characteristics of owners who wished to put in the cheapest jobs possible without much regard to the insulating qualities and the permanency of the work. It is up to the engineers to see that commercialism is not carried to a degree of foolishness.

In reply to a question of Charles Naylor as to how "Sil-O-Cel" compared with cork for cold insulation, Mr. Boeck replied that its conductivity was about 12 per cent. higher, so that as an insulator it was not quite as good, but it was not subject to dry rot or vermin, therefore for this particular use it would give satisfactory results. A number of other questions on the use and physical properties of "Sil-O-Cel" extended the meeting to a much later hour than usual and indicated the interest in the subject.

U. S. Supreme Court Decides in Water-Power Case

In a decision delivered by Justice Holmes, the Supreme Court of the United States held on Jan. 24 that the Supreme Court of Alabama was correct in not issuing a writ prohibiting the Probate Court of Tallapoosa County from taking jurisdiction of condemnation proceedings instituted by the Alabama Interstate Power Co. to take land, water and water rights belonging to the Mt. Vernon-Woodberry Cotton Duck Co. et al.

In this decision, which was heralded in various parts of the country as being revolutionary and of great import to water-power interests in general, Justice Holmes and the Supreme Court of the United States held that the purpose of the condemnation was a public one and that, as the Supreme Court could not discover anything in the decision of the Supreme Court of Alabama which ran counter to the Fourteenth Amendment, the Alabama court must be upheld. Justice Holmes declared:

The purpose of the power company's incorporation and that for which it seeks to condemn property of the plaintiff in error is to manufacture, supply and sell to the public, power produced by water as a motive force. In the organic relations of modern society it may sometimes be hard to draw the line that is supposed to limit the authority of the legislature to exercise or delegate the power of eminent domain. But to gather the streams from waste and to draw from them energy, labor without brains, and to save mankind from toil that it can be spared, is to supply what, next to intellect, is the very foundation of all our achievements and all our welfare. If that purpose is not public, we would be at a loss to say what is.

Before a corporation can condemn rights it is required to have obtained, by other means, at least an acre on each side of the stream for a damsite, and this is supposed to show that the use is not public. It is only a reasonable precaution to insure good faith. A hardly consistent argument is that the dam should be built before the necessity of taking waters below can be shown. But a plan may show the necessity beforehand. All that we decide is that no general objection based on these grounds affects the jurisdiction of the Probate Court or the constitutionality of the act.

Certain exceptions from the powers conferred, such as private residences, lands of other corporations having similar powers and cotton factories, subject to the taking of the excess of water over that in actual use or capable of use at normal stages of the stream, are too plainly reasonable so far as they come in question here to need justification. Without further discussion of the minutiæ, we are of the opinion that the Supreme Court of Alabama upon the questions arising under the Constitution of the United States was correct.

While this decision has been characterized by Governor Ammon of Colorado and other officials in close touch with water-power interests, as "far reaching and of vast importance," officials of the Supreme Court declare that its weight has been overemphasized. It is pointed out that the decision relates solely to the laws of the State of Alabama and the action of the Probate and Supreme Courts of that state. The question dealt with, it is stated, is not the actual rights of the Alabama Interstate Power Co., as much as it is the constitutionality of the Alabama laws. The Supreme Court, in this decision palpably avoided discussion of the phases of the original legal question—referring to them as "the minutiæ" of the case—and confined itself to an answer as to the relation of the Alabama laws to the Federal Constitution. In the opinion of those close to the Supreme Court, the decision might very possibly be reversed in dealing with similar conditions in other states.

Recent Court Decisions

Digested by A. L. H. STREET

Acquiring Water Rights for Power Dams—Development of hydro-electric power plants in the Southeast is resulting in considerable litigation concerning the rights of the power companies to interfere with the rights of owners of riparian lands. In one of the latest cases—Blue Ridge Interurban Railway Co. vs. Hendersonville Light and Power Co., 86 "Southeastern Reporter," 296—it is decided by the North Carolina Supreme Court that a power company that owns land on one side only of a stream is not entitled to build a dam to the middle of the stream and divert one-half the water through a flume for use in developing power, although the water be returned to the channel before leaving the company's land. In reaching this decision the court cites New York and Maine cases as upholding the principle that each of opposite riparian owners has an indivisible right to the enjoyment of the full flow of the stream. In the Georgia case of Parker vs. Central Georgia Power Co., 86 "Southeastern Reporter," 324, the Supreme Court decided that the defendant company, having acquired the right to maintain a dam, was not liable to a near-by owner of farm land on the theory of loss of crops resulting from the fact that croppers who had cultivated his land moved away on account of unhealthful conditions produced by the dam. Recovery was denied because the damages relied upon were too remote and speculative to be subject to computation. But in the suit of one Pope against the same company (page 322) it was declared by the same court that a previous release by a landowner of all claims "for damages of any nature, resulting from the construction, maintenance or operation of said dam or power plant" must be interpreted to include only claims based upon proper and nonnegligent construction and operation of the works and not as exempting the company from liability for damages negligently inflicted upon near-by landowners.

Warranty of Efficiency of Engines—When an engine is designed and sold for a specific purpose, a warranty of its capacity for work made by the seller will not be interpreted as covering use for another purpose, according to the holding of the North Carolina Supreme Court in the case of Bland vs. International Harvester Co., 86 "Southeastern Reporter," 350. The defendant sold plaintiff a threshing engine which was warranted "to be well made, of good material, and durable with proper care, and to do good work if properly operated." The plaintiff's only complaint is that plowing cannot be satisfactorily done with plows used by him, they not having been bought from the defendant. The court says on this point:

The purpose of a traction engine is to generate and furnish power. How that power is to be applied is a different matter. On the face of this contract it was contemplated that it should be used for threshing, and the plaintiff admits that it was satisfactory for that purpose. The defendant did not furnish plows nor contract that the tractor should do plowing. Whether it should be satisfactory for that purpose must depend largely upon the nature of the plows used and the method adopted for their use. This was not within the terms of the contract, and there being no allegation or proof of fraud or mistake, the court properly directed a nonsuit.

NEW PUBLICATIONS

PRINCIPLES OF DEPRECIATION. By Earl A. Saliers. Published by the Ronald Press Co., New York. Size, 8¾x6 in.; 200 pages. Price, $2.50.

Recent legislation, by forcing more systematic accounting methods, is making this subject a very live one to business men in general, but particularly to the engineer and accountant—to the former on account of his familiarity with the physical condition, depreciated value, cost of operation and replacements, and to the latter on account of his duties to properly record these and to provide funds for their replacement and the preservation of capital value.

The author, after acknowledging the somewhat unsettled condition of the subject, points out the underlying principles as now being recognized by State and Federal courts and commissions. These bodies insist that net profits must be shown clearly, and it is highly important that depreciation be duly considered and expressed in a manner that will meet with their approval; especially is this shown by the classification of the Interstate Commerce Commission as applied to railroads. The income tax in the United States and England is taken up at length, giving legal decisions and interpretations as to what can be allowed for exhaustion and limiting deductions that can be made for depreciation, repairs and renewals, and what is permitted in case of timber land and coal deposits where the principal is steadily decreasing. Our great industrial and public-utility corporations have developed public-service commissions that are going more and more into the question of adequate allowance for depreciation and cost of reproduction.

The engineer is more interested in the chapters showing the various ways for determining the depreciation charge, as the methods in common use are so confused by the point of view of individuals that there is no standard. Great accuracy is of course impossible, as obsolescence must be predicted and depends upon personal judgment. Of the various methods the simplest, known as the Straight Time Method, takes no account of interest. It does not require complicated figures, and in many applications it is the one most practical. A definite life being assumed, the same depreciation is allowed each year to reduce the purchase price to salvage value. In the Reducing Balance Method heavy depreciation is charged during the first year and a continually decreasing amount from year to year. The Sinking Fund, or Compound Interest, Method is often used without an actual sinking fund, and each year there is put aside and allowed to accumulate at compound interest during the life of the tool a fixed sum of money which will amount, at the end of the life, to the total depreciation. The Annuity Method reduces the investment at a rate equal to the increase of a sinking fund on a certain interest rate. An annuity formula with table and curve is given. The Equal Annual Payment Method is to secure a uniform investment cost, and it takes into consideration interest on the decreasing investment; it requires that depreciation follow the laws of compound interest and consequently does not appeal to many. The Unit Cost Method takes into consideration the loss in economy in running an old or obsolete machine, and therefore interests the engineer, as it is supposed to be based on the principle that the value of the old machine is such that during its life the cost of output will be the same as during the life of the new machine. This is more complicated and is inclined to underestimate the value of the old plant. As interest computations extending over long periods are more easily calculated by the use of logarithms, the book contains a chapter on this subject, while the last chapter gives a list of reference books and articles useful to the student of this subject.

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